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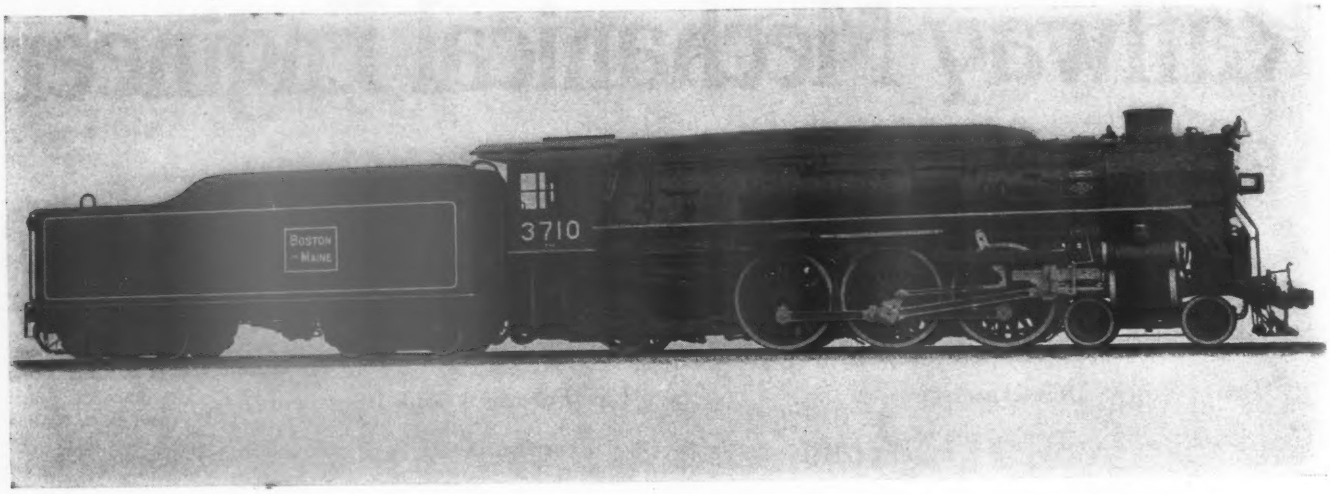
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Locomotive Axle Failures And Wheel-Press Fits[1]

By T. V. Buckwalter[2] and P. C. Paterson[3]

Research by means of fatigue tests and photo-elastic studies of models to determine location and extent of stresses due to press fits and rotative bending with means for prolonging the life of axles

ALTHOUGH it has been appreciated for some time that railroad axles have been failing because of undue concentration of stress, little has been known about how to correct the situation. In the search for an improved economic axle life the Timken Roller Bearing Company early in 1933 inaugurated a research program to learn, if possible, how to avoid axle failures which had been occurring at stresses below the usual safe working range. Fatigue failures occurring just inside the face of press fitted members will be discussed in this paper. The program carried on at the University of Michigan and the Timken physical laboratory included both physical tests and photo-elastic studies and during the research much valuable data were gathered.

Application of the usual formulas for determining the press-fit pressure between two cylinders gives a uniform pressure distribution when the cylinders are both of the same length. In the case of a sleeve or wheel pressed on an axle, however, the press-fit stresses are no longer uniform, but increase rapidly near the end faces of the sleeve due to the greater compressive resistance of the protruding portions of the axle. This unfavorable stress condition is further complicated by the fact that the axle rotates under bending stresses—tension on the top and compression on the bottom side.

The end faces of the sleeve have a tendency to impinge against the compression side of the axle. As the axle rotates the alternate contraction and elongation of the axle fibers produces a minute sliding action of these faces on the axle. This periodic action produces a band of abraded axle metal near the end faces of the sleeve which is very noticeable in the usual press-fit failures and is often accompanied by a brown rust formation which introduces a further weakening effect known as corrosion fatigue.

Physical Tests

Data pertaining to physical tests on carbon-steel axles are presented in Fig. 1. Although the work is not yet complete, the data now available tend to favor (a) raised seat on the axle; (b) stress relief groove in wheel hubs, and (c) cold working of wheel seats, as prolonging the life of axles. For example, some test specimens embodying these features have already shown 40 times the life of normal carbon-steel plain axles. It should be noted that the possibilities of improving axle life by means of cold working the wheel seats is recognized as an outstanding development, both in this country and abroad, and is already the subject of intensive research on the Timken program.

The data derived from tests on 64 axles indicate an improvement of 25 per cent in fatigue strength of plain carbon steel as used in locomotive axles without cold working. Thus an axle having a normal life of 400,000 miles under present-day conditions might be given a life of at least 1,500,000 miles by comparatively minor modifications. However, if the axle is cold worked in addition to being provided with raised seats and stress-relief grooves in the wheel hubs, it is within the realm of possibility that practically the full fatigue strength of a normalized steel axle (39,000 lb. per sq. in.) can be made available for design purposes.

The use of alloy steels when properly heat treated offers opportunities for marked improvement in axle life, but unless the alloy steel is properly heat treated there is no advantage whatsoever over carbon steel.

The physical tests, made on 2-in. diameter shafts, were conducted on cantilever type rotating beam machines. Loads were applied to the specimens through double-

[1] Abstract of a paper presented before the Engineers Society of Western Pennsylvania, January 29, 1935.
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row Timken bearings mounted in self-alining housings. The press-fitted members were applied to the specimens .0005 in. tight per inch of diameter. The machines were operated continuously at 2,000 r.p.m. until failure occurred.

Photo-elastic Study of Press-Fitted Members

A qualitative analysis of a series of photo-elastic studies showing the stress concentration developed in various press-fitted assemblies, with particular reference to driving, trailer and tender axles, is herewith presented, discussing in detail

(a) equipment
(b) character of photo-elastic fringes on an axle not subject to press-fit stresses
(c) character of photo-elastic stresses on integral shoulders and press-fit members
(d) character of fringes at fillets on wheel hubs
(e) photo-elastic fringes with press-fit members, such as driving or tender wheels, having stress-relief grooves on the inner face of the hub.

These photo-elastic studies show that the strength of an axle is considerably weakened by the conventional press fit. This checks with service and fatigue tests as to where axle failures occur. Means of improving this stress concentration are indicated by the tests of various shaped grooves on the inside face of the wheel hub and by the use of a raised wheel seat.

The following conclusions may be drawn from this photo-elastic study:

1—The conventional type of press-fit assembly of a plain wheel hub on a uniform diameter axle, Fig. 6, gives a stress concentration of about the same magnitude as a very small fillet; in this comparison the hub is considered as being machined as an integral part of the axle, as in Fig. 5.

2—The effect of a radius at the inside hub face of the wheel, Fig. 6, offers no noticeable improvement over a sharp corner on the wheel hub, Fig. 7.

3—A wheel hub mounted against a shoulder on an axle, Fig. 8, gives a weaker axle construction than the conventional assembly.

4—A grooved wheel hub mounted on a uniform diameter axle, Fig. 9, reduces the stress concentration considerably over the conventional assembly.

5—Of the six types of relief grooves tested type 5, Fig. 10 (also photo-elastic studies Figs. 9 and 12) gives the least stress concentration. The other types of grooves tested, which differed in shape, depth and location, also show improvements over the conventional assembly.

6—A plain type of wheel hub mounted on an axle with a raised wheel seat or pad, Fig. 11, gives an improvement in stress concentration over the conventional assembly.

7—A certain height of pad gives maximum reduction in stress concentration and the use of a higher pad does not give any further beneficial effect.

Designs Tested and Equipment Used

The axles studied were of the conventional outboard type. The Bakelite test models were designed to show stress concentration and distribution as affected by modifications of the inside face of the wheel hub and by changes in wheel-seat diameter.

All axles studied had a journal diameter of 8½ in., but the wheel-seat diameter varied from 9¾ in. to 12 in. The wheel-hub sections at the inside hub face were approximately 4 in. thick. Various wheel-hub modifications were tested also. All models were made of Bakelite approximately ⅜ in. thick and were one-sixth

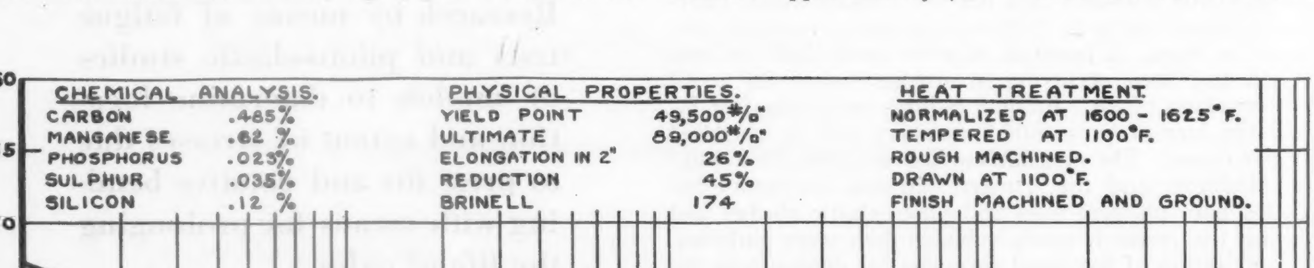


Fig. 1—Fatigue test data of carbon-steel axles, normalized and tempered—Specimens 2 in. diameter; press fit of hub, .001 in. tight

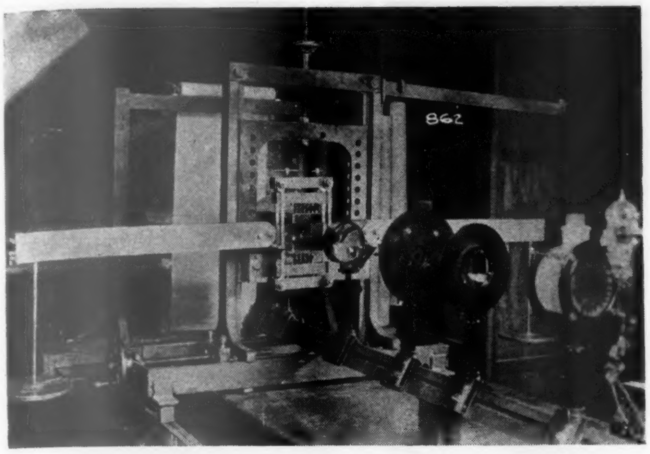


Fig. 2—General view of photo-elastic equipment and model in position

of the actual dimensions of the wheel-hub and axle assembly.

Method of Loading and Photo-elastic Set-Up

This is shown in Figs. 2 and 3. Fig. 3 indicates the method of simulating the press-fit condition, calibrated springs pressing the hub against the axle. Fig. 2 shows a general view of the complete axle and wheel model under load in position in the photo-elastic set-up. Long steel beams carrying pans are fixed to the ends of the axle model. This construction permits pure bending stresses to be applied to the model so that simultaneous bending and press-fit stresses may be obtained.

A photo-elastic study showing fringes for the case of a uniform beam with pure bending stresses only is shown in Fig. 4. Parallelism and equal spacing of fringes indicate uniform stress distribution. Fringe marked *0* is the neutral axis. Shearing stresses shown are directly proportional to fringe order marked on each fringe.

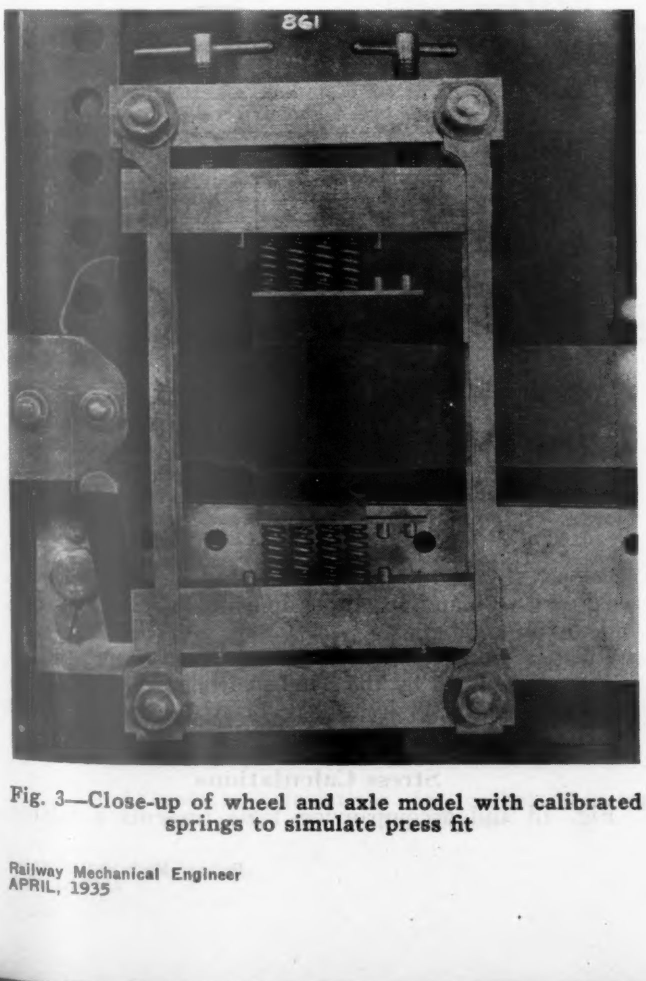


Fig. 3—Close-up of wheel and axle model with calibrated springs to simulate press fit

Another study of bending stresses is shown in Fig. 5. This covers the portion of an axle enlarged to represent a wheel hub which, in this case, is studied as an integral part of the axle. Note the crowded and sharply curving fringes at the very small fillets adjoining the enlarged portion, which indicates localized stress and the point where failure will occur. This integral study shows that the conventional press fit gives a stress concentration of about the same magnitude as a very small fillet or sharp corner. Compare this with Fig. 6.

Discussion

A conventional press fit of a plain wheel hub on a uniform diameter axle, Fig. 6, gives peak press-fit stresses at the end faces of the wheel hub. It should be noted that the maximum bending stresses also occur at the same position on the axle. The bending stresses are much greater than those normally calculated due to the abrupt change in section from the axle into the hub. It is evident, therefore, that both these unfavorable conditions occur at the same location on the axle, thereby creating an accumulative weakening effect. If in the case where a wheel hub is mounted against a shoulder on an axle, Fig. 8, the hub radius is made to fit the small shoulder radius; then, in addition to the weakening effect mentioned above there exists the unfavorable effect of the stress concentration of the fillet occurring at this same critical section. This type of shoulder construction

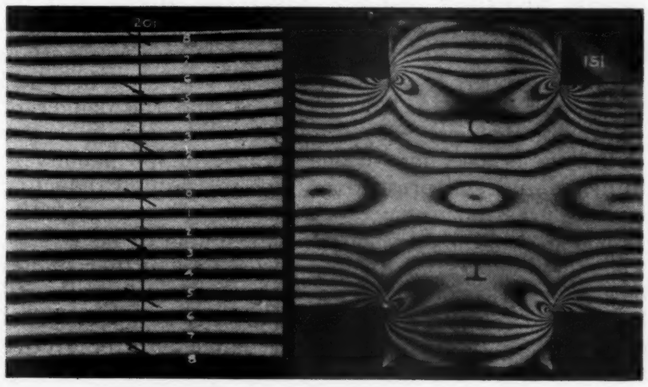


Fig. 4 (Left)—Fringes in uniform beam under bending stresses; Fig. 5 (Right)—Fringes in beam with enlarged portion representing wheel hub when under bending stresses

weakens the axle, but its effect is sometimes small as compared to the total stress concentration present.

A conventional press fit of a plain wheel hub on an axle with raised wheel seat or pad, Fig. 11, has three beneficial effects: A more gradual change in section takes place between the axle and the wheel hub; the peak pressure at the end face of the wheel hub is reduced, due to the decrease in lateral restraint of the protruding axle, and the bending stresses in the axle near the end of the pad are reduced at the point where the press-fit stresses are a maximum.

A grooved wheel hub on a uniform diameter axle, Fig. 9, presents a very favorable fringe pattern, showing an improvement over the cases discussed above. Grooving has the effect of relieving the peak press-fit stresses near the end face of the hub. This is due to the flexibility of the lip on the hub which prevents the building up of high stresses at this point. Grooving also effects a gradual transition in section from the axle into the sleeve, functioning in the manner of a large fillet.

From the standpoint of economical design, grooving

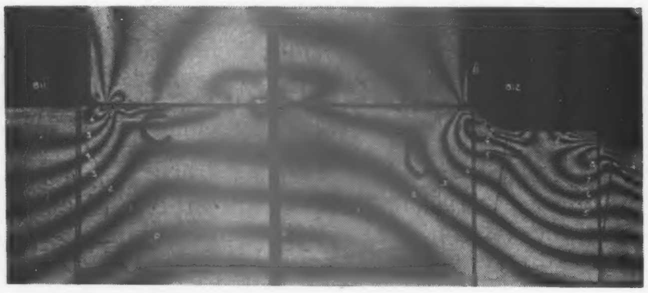


Fig. 6—Straight axle seat with hub assembly having a ⅛-in. radius on the inner face. Stress concentration near the inner hub face corresponds with the zone of failure in service

appears to offer improvement over the use of a plain hub mounted on a pad on the axle. The axle forging design is simplified and space requirements are reduced.

The combination of a raised wheel seat or pad with a grooved wheel hub, Fig. 12, gives a very favorable fringe pattern. This combination may be expected to show some improvement over all other designs. Preliminary analysis of the Timken photo-elastic studies, however, does not show definitely how much improvement is represented by this combination over the case where a grooved hub is applied on a straight axle, Fig. 9. This question is being given further analysis.

Six different types of grooves were tested. It is natural to expect that the larger the groove and the thinner the wedge-shaped lip the greater will be the improvement in stress concentration. There are certain limits to groove formation, such as reduction in wheel tonnage, reboring of wheel centers, and the danger of failure of material with thin sections at the lip. Too thin a lip is also undesirable because of its inability to transmit sufficient pressure to give the desired reduction in peak press-fit pressure.

The best groove formation was found to be type 5, Figs. 9, 10 and 12, and it is expected that further studies now under way will indicate that the pressure peaks will be reduced for this type over the other grooves. This groove shape has a 20-deg. angle tangent to the groove radius and gave better results than either the 30-deg. angle or the 45-deg. angle or any of the other groove formations.

Various heights of wheel-seat pads were investigated. It is apparent that some particular height of pad gives maximum reduction in stress concentration and that the use of a higher pad does not give any further appreciable beneficial effect. This phase is being given further study.

A two-dimensional stress distribution is indicated in these photo-elastic studies and the question may arise

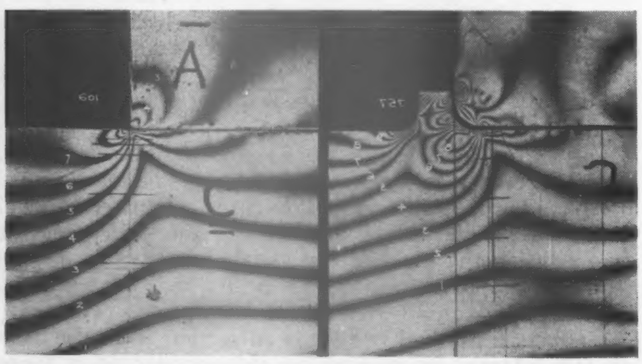


Fig. 7 (Left)—Straight axle and hub with sharp corner; Fig. 8 (Right)—Axle with hub mounted against a shoulder

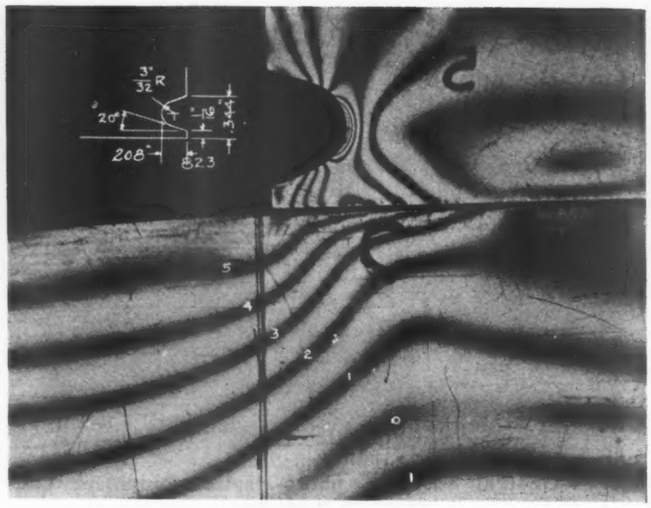


Fig. 9—Straight axle with grooved wheel hub assembly, type 5—Note the reduction in axle stress concentration

as to the applicability of this method in view of the three-dimensional case actually present in the press-fit assembly. However, these photo-elastic studies have their value from a qualitative standpoint. They permit a comparison of one fringe pattern with another; determine the stress concentration due to shape; give an indication of the distribution of pressure across the length of the press fit; and develop a broader aspect of the entire problem. With this in mind fatigue tests can be planned more directly toward a definite design, thereby eliminating the necessity for determining certain variables from a series of fatigue tests requiring considerable time and expense. It is not believed that definite numerical data can be obtained on stress concentration from photo-elastic studies. It is necessary to resort to fatigue tests in order to determine the actual stress concentration as

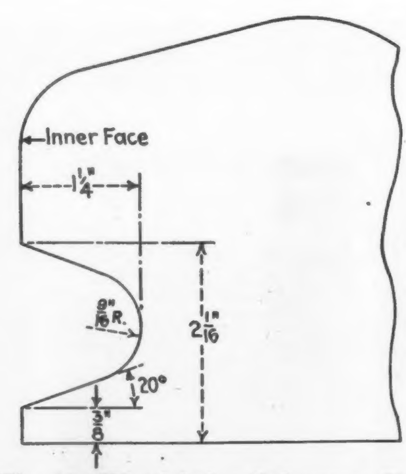


Fig. 10—Wheel hub relief groove No. 5 which showed best improvement in reduced axle stresses

developed by the combination of the effects of size, sensitivity of materials, three-dimensional stress system, and corrosion fatigue.

Further work using a sensitive lateral extensometer is being continued by the Timken Roller Bearing Company at the University of Michigan to determine the principal stresses.

Stress Calculations

Fig. 13 and accompanying table presents a tabular

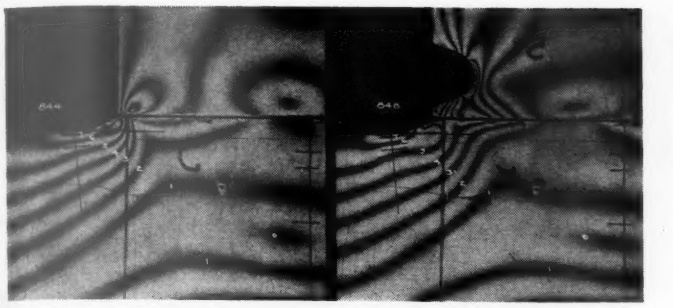


Fig. 11 (Left)—Raised seat on the axle with a hub having a sharp corner; Fig. 12 (Right)—Raised seat on the axle with a hub having relief groove—Note the difference in concentrated axle stresses

analysis of stresses occurring in a typical outboard-bearing railroad axle. It is particularly interesting to note that the maximum shear stress at the inside hub face using a stress relief groove in the hub is only 5,400 lb. per sq. in. as compared with a stress of 13,700 lb. per sq. in. when a plain type hub is used.

A comparison of the Timken type axle, which embodies the features discussed in this paper, with the standard A.R.A. type E axle, is shown in Fig. 14. The Timken axle is shown above the center line, while the A.R.A. axle is indicated below.

The A.R.A. axle has three concentrating factors consisting of (a) the presence of a shoulder on the portion of the axle subjected to high bending stress; (b) use of a comparatively thick hub without relief, building up higher stress concentration at the face of the hub, and (c) the superimposing of (a) and (b), producing maximum stress concentration at the inner hub face (illustrated in worst possible form in Fig. 8).

Railroad axles for outboard-bearing engine trucks,

Summary of Stresses — 7 5/8" Axle

Line No.		Section at Outside Face of Hub "A-A"	Section at ℄ of Rail "B-B"	Section at ℄ of Hub "C-C"	Section at Inside Face of Hub without Relief "D-D"	Section at Inside Face of Hub with Relief "D-D"	Section at Point 21" from ℄ of Axle "E-E"	Section at ℄ of Axle "F-F"
1	Bending from Rail Reaction	4700	13800	13400	11500	11500	15000	15700
2	Direct Compression from Thrust Reaction	—	—	-500	-500	-500	-500	-700
3	Net Bending Stress	4700	13800	12900	11000	11000	14500	15000
4	Radial Hoop Stress	12300	12300	14800	12300	4000	—	—
5	Tangential Hoop Stress	12300	12300	14800	12300	4000	—	—
6	Crushing Stress	3400	2400	—	3400	3400	—	—
7	Direct Shear from Rail Reaction	-700	—	—	-700	-700	—	—
8	Maximum Shear	13700	13500	14800	13700	5400	—	—
9	Maximum Combined Tension	16300	22100	22900	20300	14300	14500	15000

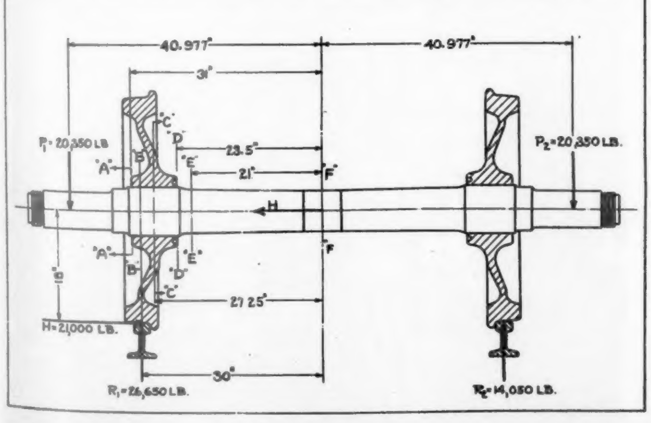


Fig. 13—Axle-stress calculations for outboard-bearing axle under the loading shown

drivers, outboard trailers, tenders, and freight and passenger cars are subject to improvement by using the Timken construction shown above the center line in Fig. 14 which eliminates the integral collar, introduces a stress relief groove, and provides a raised seat for the press fit of the wheel hub. There is also definite probability of improvement in axle service by cold working of the wheel seat.

Improvement in axle service could be obtained at slight expense in rolled wheels by providing an extended

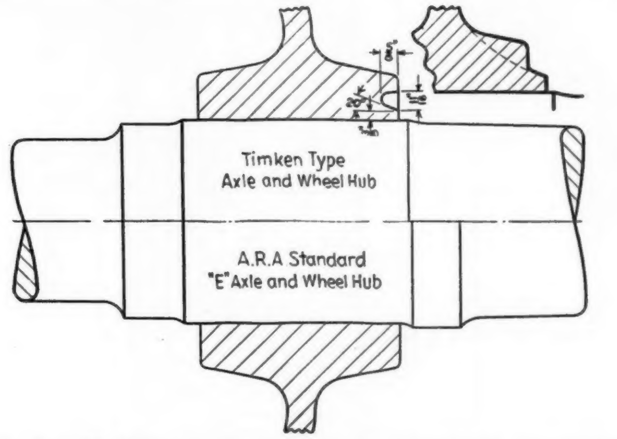


Fig. 14—Timken improved type axle (above) compared with A. R. A. standard axle (below)

hub of reduced diameter corresponding with the 20-deg. lip on the relief groove, as shown in Fig. 14.

The relief groove on the outer wheel hub face is not considered necessary because the stress in the axle is greatly reduced at that point, that portion of the axle not being subject to stress increase arising from flange action.

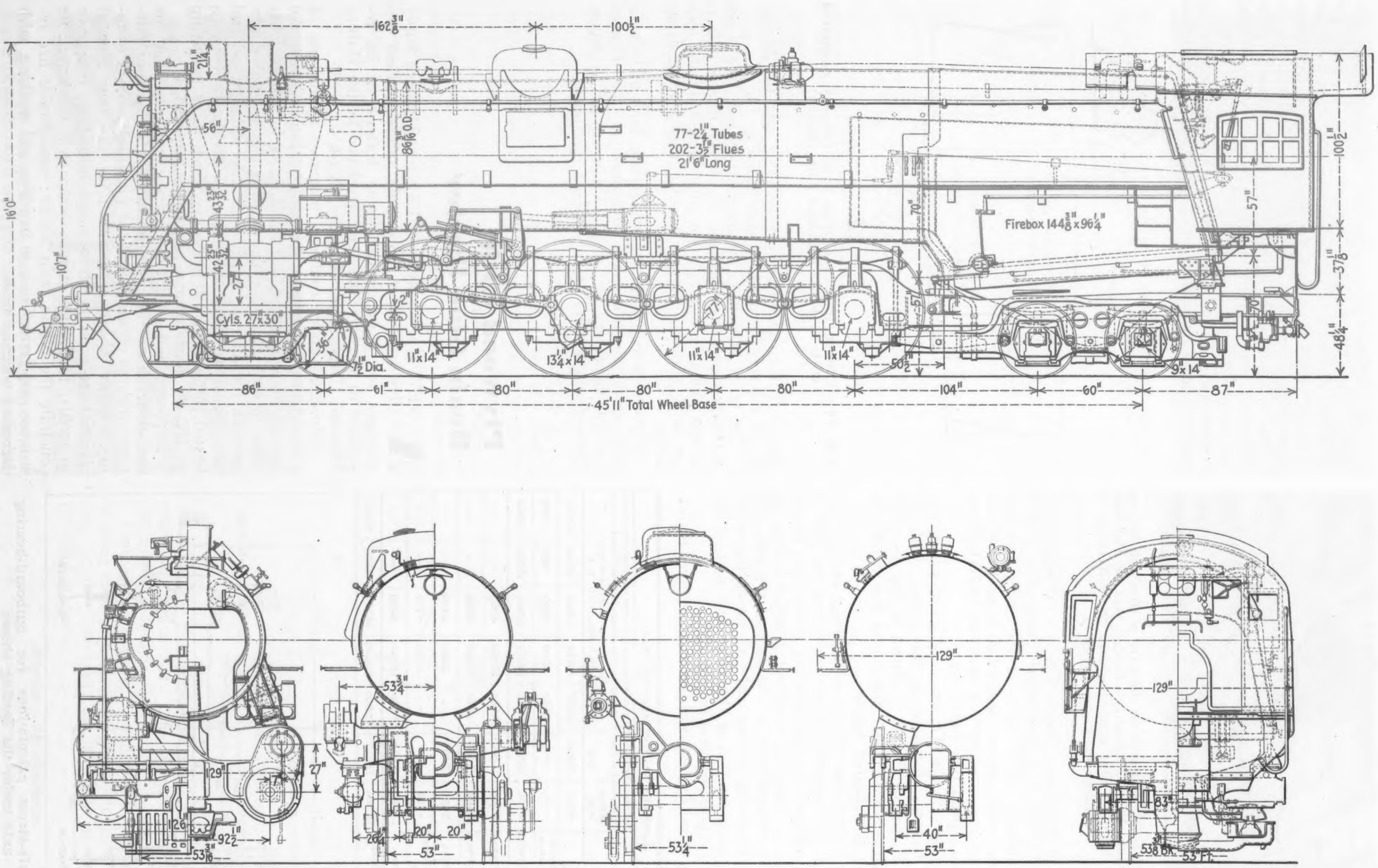
Physical tests for determining fatigue limit of reduced axle specimens are being continued at the Timken physical laboratory and at the University of Michigan. Photo-elastic studies are also being continued.

Plywood with Resin Glue Binder

A NEW type of plywood, known as Harbord Super Plywood, which apparently offers marked advantages for passenger-car lining because of its resistance to moisture, has been developed by the Harbor Plywood Corporation, Hoquiam, Wash.

Fabricated under exclusive processes by the use of a new type of resin glue invented by James Nevin, the new plywood is glued, dried and ready for use in from three to five minutes. The resin binder is applied dry between the alternating plies of fir or other wood and each panel is then hot-pressed individually between heated plates in a 350-ton press which takes panels up to 102 in. in width and as long as required.

Exposure and boiling tests have indicated that the new material can be used indefinitely for outdoor purposes with no separation of the plies and without warping. Other advantages claimed for this new type of plywood are great flexibility; high insulation; resistance to molds, fungi and insects, which will not attack or penetrate the Nevin resin binder; considerable fire resistance and greater ease in handling and working, with decreased wear on tools.



Elevation and cross sections of the 4-8-4 type locomotives, Class T-3, built by the Baldwin Locomotive Works for the Lehigh Valley

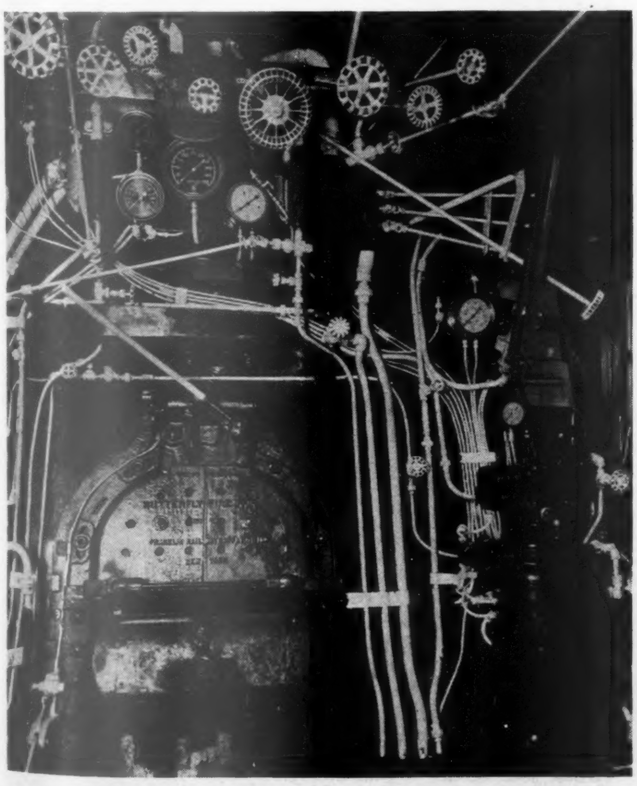
Additional 4-8-4 Heavy Fast Locomotives for the Lehigh Valley

Five more 4-8-4 type locomotives bring total for Class T up to 27—New power has larger drivers, higher boiler pressure and increased grate area

EARLY in 1931 the Lehigh Valley placed orders for two sample 4-8-4 type locomotives, one each with the Baldwin Locomotive Works and the American Locomotive Company. These locomotives were designed to handle fast freight trains of 3,000 tons between Buffalo and tidewater. A specified schedule covering train movements was drawn up by the road and given to the builders. On comparatively level sections the running time specified over a division was at a rate of 30 m.p.h. Helper service was to be used at only one point.

A general description of the two sample locomotives—No. 5100, road class T-1, and No. 5200, road class T-2—was given in the *Railway Mechanical Engineer,* May, 1931, page 237. Before being placed in regular service they were submitted to full dynamometer road tests. Both locomotives satisfactorily handled trains somewhat in excess of the specified 3,000 tons and in less than the time called for. Tests showed indicated horsepowers as high as 4,100; drawbar horsepowers up to 3,865 at about 35 m.p.h.; coal consumption from 3.16 to 3.55 lb. per drawbar horsepower hour; an evaporation of 6.32 to 6.92 lb. of water per pound of coal; 19.9 to 23.8 lb. of water per drawbar horsepower hour, and an average mechanical efficiency of about 92 per cent.

Up to 1931 Lehigh Valley fast freight trains were handled by Class K-5½ Pacific type locomotives with 73-in. drivers, weighing 311,900 lb. and having a rated tractive force of 48,723 lb., or by class N-5 Mikado type locomotives with 63-in. drivers, weighing 325,000 lb. and having a rated tractive force of 63,000 lb. The Mikados were also equipped with boosters having a tractive force of 11,000 lb.



Right-hand side of the back boiler head in the cab of the Lehigh Valley locomotive

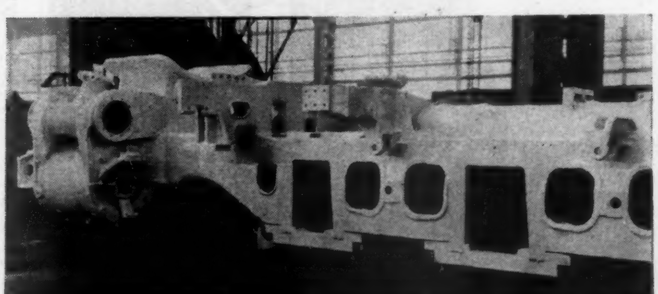
The sample locomotives, having proved so satisfactory, an additional order for 10 locomotives were given to each of the two builders, delivery being made in the spring of 1932. Practically no changes were made in either design, but somewhat larger tenders were specified. While the two sample locomotives had tenders with a capacity for 18,000 gallons of water and 28 tons of coal, subsequent locomotives were provided with tenders having a capacity for 20,000 gallons of water and 30 tons of coal.

Table 1—General Comparison of Lehigh Valley 4-8-4 Type Locomotives

Road class	T-1	T-2	T-3
Road number	5101	5201	5126
Builder	Baldwin	American	Baldwin
Weight, engine, lb.	413,170	424,000	435,000
Weight on drivers, lb.	270,000	269,000	272,200
Rated tractive force, engine, lb.	66,400	66,700	66,500
Tractive force, aux. loco., lb.	18,360	18,360	....
Cylinders	27 in. by 30 in.	26 in. by 32 in.	27 in. by 30 in.
Drivers, in.	70	70	77
Steam pressure, lb.	250	255	275
Heating surfaces, sq. ft.:			
Firebox, complete	490	508	507
Tubes and flues	4,932	4,933	4,932
Evaporative	5,422	5,411	5,439
Superheating	2,256	2,243	2,056
Grate area, sq. ft.	88	88.3	96.5

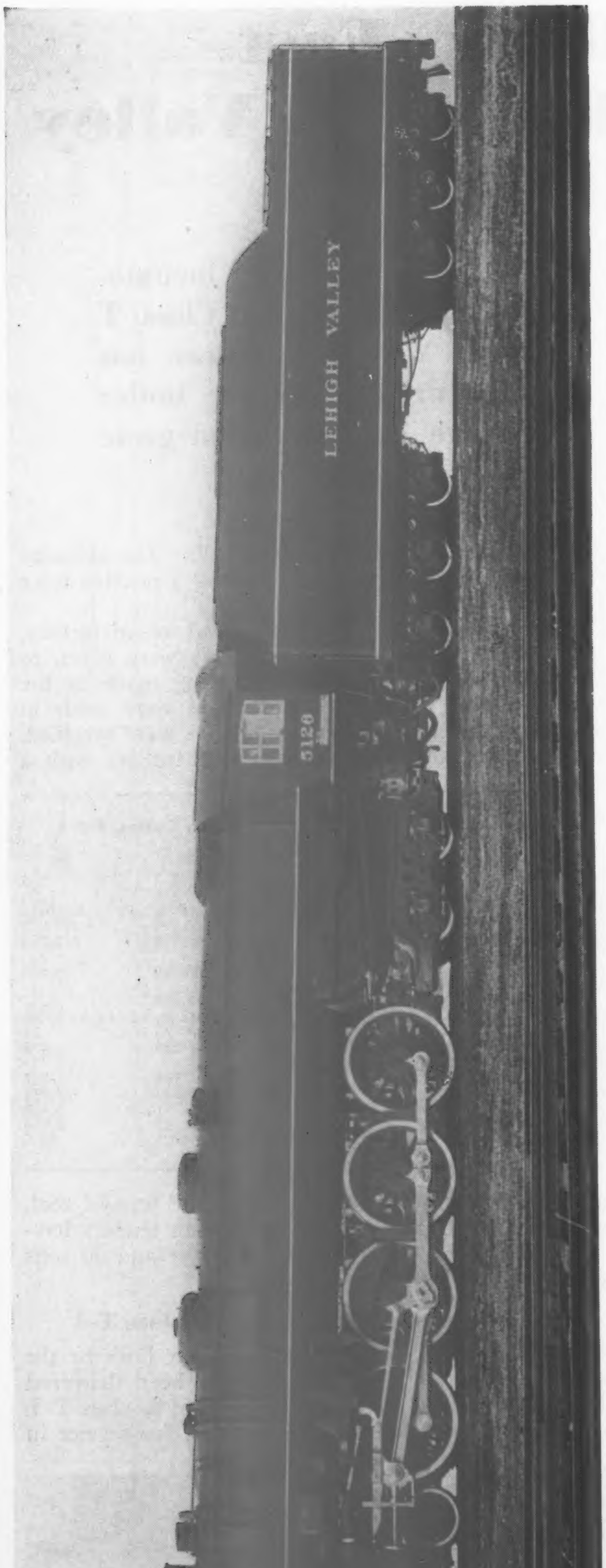
New Locomotives Designated as Class T-3

Five more locomotives of the 4-8-4 type, built by the Baldwin Locomotive Works, have now been delivered to the Lehigh Valley. They are designated as class T-3, numbered 5125 to 5129, and are intended for service in



Front of cast-steel bed for Lehigh Valley 4-8-4 type locomotives—it includes air reservoir and cylinders with integral back heads

either fast freight or heavy passenger runs between Jersey City, N. J., and Buffalo, N. Y. In general design they are a development of the previous class T-1 and T-2 engines. A general comparison of the leading weights and dimensions of the three locomotive classes is given in one of the accompanying tables. It will be noted that in the new class T-3 the diameter of the drivers



Lehigh Valley 4-8-4 type locomotive, Class T-3, designed for fast freight and heavy passenger traffic—Built by the Baldwin Locomotive Works

Table 11—Principal Dimensions, Weights and Proportions of the Lehigh Valley 4-8-4 Type Locomotives, Class T-3

Railroad	Lehigh Valley
Builder	Baldwin Locomotive Wks.
Road Nos.	5125-5127
Road class	T-3
Type of locomotive	4-8-4
Service	Freight and passenger
Height to top of stack	16 ft. 0 in.
Width	11 ft. 0 in.
Cylinders, diameter and stroke	2—27 in. by 30 in.
Valve gear, type	Baker
Valves, piston type:	
Size	12 in.
Maximum travel	8⅝ in.
Steam lap	1⅝ in.
Exhaust clearance	$^3/_{16}$ in.
Lead in full gear	¼ in.
Cut-off in full gear, per cent	82½
Crank throw	25 in.
Weights in working order:	
On drivers	272,200 lb.
On front truck	73,700 lb.
On trailing truck	89,100 lb.
Total engine	435,000 lb.
Tender	389,000 lb.
Wheel bases:	
Driving	20 ft. 0 in.
Rigid	13 ft. 4 in.
Total engine	45 ft. 11 in.
Total engine and tender	95 ft. 2 in.
Wheels, diameter outside tires:	
Driving	77 in.
Front truck	36 in.
Trailing truck	42 in.
Journals, diameter and length:	
Driving, main	13¼ in. by 14 in.
Driving, others	11 in. by 14 in.
Front truck	7½ in. R. B.
Trailing truck	9 in. by 14 in.
Boiler:	
Type	Conical
Steam pressure	275 lb.
Fuel	Soft coal
Stoker	Standard BK
Diameter, first ring, outside	$86^3/_{16}$ in.
Diameter, third course, outside	98¼ in.
Firebox, length and width	144⅛ in. by 96¼ in.
Height mud ring to crown sheet, back	72⅝ in.
Height mud ring to crown sheet, front	91⅜ in.
Combustion chamber length	48 in.
Tubes, number and diameter	77—2¼ in.
Flues, number and diameter	202—3½ in.
Length over tube sheets	21 ft. 6 in.
Grate type	Firebar
Grate area	96.5 sq. ft.
Stoker	Standard BK
Heating surfaces:	
Firebox and comb. chamber	364 sq. ft.
Syphons (4)	143 sq. ft.
Firebox, total	507 sq. ft.
Tubes and flues	4,932 sq. ft.
Total evaporative	5,439 sq. ft.
Superheating (Type E)	2,056 sq. ft.
Comb. evaporative and superheating	7,495 sq. ft.
Feedwater heater	Worthington—5SA
Tender:	
Style	Rectangular W.B.
Water capacity	20,000 gal.
Fuel capacity	30 tons
Trucks	6-wheel
General data estimated:	
Rated tractive force, 82½ per cent cut-off	66,500 lb.
Potential horsepower (Cook)	4,332
Speed at 1,000 ft. piston speed	45.85 m.p.h.
Piston speed at 10 m.p.h.	218.3 ft. per min.
Weight proportions:	
Weight on drivers ÷ total weight engine, per cent	62.6
Weight on drivers ÷ tractive force	4.09
Weight engine ÷ comb. heat. surface	58.1
Boiler proportions:	
Tractive force ÷ comb. heat. surface	8.87
Tractive force × dia. drivers ÷ comb. heat. surface	683
Combined heat. surface ÷ grate area	77.7
Firebox heat. surface ÷ grate area	5.26
Firebox heat. surface, per cent of evap. heat. surface	9.32
Superheat. surface, per cent comb. heat. surface	27.4

Table III—Special Equipment Applied on Lehigh Valley 4-8-4 Type Locomotives, Class T-3

Builder	Baldwin Locomotive Works
Engine numbers	5125—5129
Air brakes, 8-ET	Westinghouse Air Brake Co.
Air compressors—2-8½ in. c.c.	Westinghouse Air Brake Co.
Arch, brick, Security	American Arch Co.
Axles, driving	Standard Steel Works Co.
Axles, engine truck	Standard Steel Works Co.
Axles, tender	Bethlehem Steel Co.
Bearings, roller—Front truck (5)	Timken Roller Bearing Co.
Bearings, roller—Driver, trailer, tender (2)	Timken Roller Bearing Co.
Bed, locomotive	General Steel Castings Corp.
Bell ringer	Superior Ry. Products
Blower nozzle	T-Z Ry. Equip. Co.
Blower valve	Lunkenheimer Co.
Blow-off valve	Wilson Engineering Co.
Brakes, driver	American Brake Co.
Brakes, tender—clasp	American Steel Foundries
Brake shoes (driver-tender)	American Brake Shoe & Fdry. Co.
Buffer, radial	Franklin Ry. Supply Co.
Bushings, cylinder and valve	Hunt-Spiller Mfg. Corp.
Cab windows—clear vision	Prime Mfg. Co.
Check valves, feedwater heater	Consolidated Ashcroft Hancock Co.
Check valve, injector	Consolidated Ashcroft Hancock Co.
Couplers, engine	National Malleable & Steel Cast. Co.
Coupler, tender	Gould Coupler Co.
Cylinder cocks	Ardco Mfg. Co.
Cylinder relief valves	Walworth Mfg. Co.
Draft gear—Type A-94-XB	W. H. Miner
Draft-gear attachments	Symington Co.
Drawbar—Unit Safety	Franklin Ry. Supply Co.
Drifting valve steam line check	Lunkenheimer Co.
Drip cocks, dynamo	Lunkenheimer Co.
Driving-box brasses	Magnus Metal Co.
Driving-box grease cellar	Franklin Ry. Supply Co.
Driving-wheel wearing plates	Magnus Metal Co.
Dust guards, tender	Cottman Co.
Dust guard closures	Cottman Co.
Feedwater heater—Type 5-S-A	Worthington Pump & Mfg. Corp.
Fire door—Butterfly No. 8	Franklin Ry. Supply Co.
Flexible connections, engine and tender	Barco Mfg. Co.
Flue blower	Superior Ry. Products Corp.
Driving-box safety bombs (2)	Timken Roller Bearing Co.
Frame, tender—water bottom	General Steel Castings Corp.
Gages, steam	Consolidated Ashcroft Hancock Co.
Grates—firebar	Waugh Equip. Co.
Headlight	Pyle-National Co.
Headlight generator	Sunbeam Electric Mfg. Co.
Heating equipment reducing valve	Leslie Co.
Injector, live steam—Type LNL	Consolidated Ashcroft Hancock Co.
Lagging	Johns-Manville Sales Corp.
Lamps, cab	Pyle National Co.
Lamps, classification	Handlan-Buck Mfg. Co.
Lateral motion driving box	Franklin Ry. Supply Co.
Lubricator—Bosch (air compressor)	Q & C Co.
Lubricator, mechanical	Detroit Lubricator Co.
Lubrication—Alemite fittings	Prime Mfg. Co.
Lubrication fittings, driving-box faces (grea e)	Universal Lubricating Systems, Inc.
Lubrication hose, engine and tender, stoker trough	Packless Metal Products Co.
Netting, smokebox—Draftax No. 393	W. S. Tyler Co.
Packing, piston and bull ring	Hunt-Spiller Mfg. Corp.
Packing, piston rod and valve stem	U. S. Metallic Pack. Co., King type
Pilot (cast steel)	General Steel Castings Corp.
Pipe (wrought iron) (3)	A. M. Byers Co.
Pipe (wrought iron) (2)	Reading Iron Co.
Plugs, arch tube	American Locomotive Co.
Plugs, washout	American Locomotive Co.
Reverse gear	Franklin Ry. Supply Co.
Rods	Standard Steel Works Co.
Rod bushings, bronze	Magnus Metal Co.
Rod bushings, connecting rod stub	Hunt-Spiller Mfg. Corp.
Safety valves	Consolidated Ashcroft Hancock Co.
Sanders	U. S. Metallic Pack. Co.
Side bearings, tender	Edwin S. Woods & Company
Smokebox hinges	Okadee Co.
Springs, driving, engine truck, trailer, tender	Ry. Steel Spring Co.
Sprinkler, coal	Consolidated Ashcroft Hancock Co.
Staybolts, flexible	Flannery Bolt Co.
Steel sheets, boiler and firebox—nickel steel	Bethlehem Steel Co.
Stoker—Type BK	Standard Stoker Co.
Superheater—Elesco Type E	Superheater Co.
Syphons, Thermic	Locomotive Firebox Co.
Tank hose couplings	Consolidated Ashcroft Hancock Co.
Tank valves	Everlasting Valve Co.
Tender-truck boxes	Symington Metal Co.
Tender-truck brasses	Magnus Metal Co.
Tender frame	General Steel Castings Corp.
Throttle valve—Bradford	American Throttle Co.
Tires, driving and trailer	Midvale Steel Co.
Train control, automatic	General Ry. Signal Co.
Truck, engine (frame)	General Steel Castings Corp.
Truck, trailing (frame)	General Steel Castings Corp.
Truck, tender	General Steel Castings Corp.
Tubes and flues, steel	National Tube Co.
Valves, cab	Walworth Co.
Valve gear—Baker	Pilliod Co.
Valves, feedwater-heater throttle	Consolidated Ashcroft Hancock Co.
Valves, stoker	Lunkenheimer Co.
Water gage	Nathan Mfg. Co.
Water column	Nathan Mfg. Co.
Wedges, adjustable	Franklin Ry. Supply Co.
Wheels, engine truck	Standard Steel Works Co.
Wheels, trailer truck	Standard Steel Works Co.
Wheels, tender	Standard Steel Works Co.
Whistle	Consolidated Ashcroft Hancock Co.
Whistle operator	Viloco Ry. Equip. Co.

has been increased from 70 in. to 77 in. and the boiler pressure raised from 250 to 275 lb. As a result the rated tractive force has remained practically constant while the proportions make the new locomotives better adapted for still greater speeds in harmony with the traffic demands at the present time. The firebox was made a foot longer, the weights were somewhat increased and the wheel bases extended. In keeping with general road restrictions for through service the stack height is held to 16 ft. and the width to 11 ft. The locomotives are designed, furthermore, to operate on 1½ per cent grades and to pass curves of 18 deg.

Boiler and Boiler Mountings

The boiler is of the conical type and carries 275 lb. steam pressure. The outside diameter of the first ring is 86$\frac{3}{16}$ in. and the largest outside diameter at the third course is 98¼ in. Nickel steel sheets, furnished by the Bethlehem Steel Company, are employed for both shell and firebox. The firebox crown and side sheets are in three pieces, which, in addition to the door sheet throat sheet, flue sheet and combustion chamber, are completely welded. The firebox is 144⅜ in. long and 96¼ in. wide, which provides a grate area of 96.5 sq. ft. Three Thermic syphons on which the Security brich arch is carried are fitted in the firebox and an additional syphon is located in the combustion chamber which is 48 in. long. The grates are of the Firebar type and the coal is fed by a Standard Type BK stoker with a cast-steel trough on the tender. The superheater is a Type E, having 2,056 sq. ft. of surface. Auxiliaries operated by superheated steam are the whistle, drifting valve and blower valve.

The feedwater heater is a Worthington, type 5-S-A and, for supplementary boiler feeding, a Hancock non-lifting injector, type LNL, is fitted. The feedwater check valves are located on top of the boiler near the front end.

Three safety valves of the Consolidated type are located a short distance back of the dome and are attached directly to the boiler shell which is suitably reinforced at this point. Superior flue blowers are provided. The whistle, controlled by a Viloco whistle operator, is mounted on the side of the smokebox in an inclined position. Blow-off valves and mufflers are of the Wilson type. Sanders were furnished by the U. S. Metallic Packing Company and the boiler lagging by Johns-Manville.

Foundation and Running Gear

The locomotive is built on a one-piece cast-steel bed furnished by the General Steel Castings Corporation. The bed casting includes the cylinders with integral back heads, an air reservoir and various brackets, including those for the air compressors on the front deck. Its shipping weight was approximately 66,500 lb. Pilots were not fitted on the previous class T-1 and T-2 locomotives, but have been provided on the new class T-3 locomotives. The engine coupler is counterbalanced and pivoted so that it can swing down clear into the cast-steel pilot when not in use. All drivers and the trailing truck are equalized together by means of a transverse lever back of the rear drivers which is connected to the trailer-truck equalizers.

(*Continued on page 143*)

Light-Weight Motor Cars On the Norfolk Southern

The "Rail Bus" is designed for single-unit operation. The 176-hp. gas engine provides a high horsepower-weight ratio for the 41,000-lb. vehicle which seats 53 passengers and has a small baggage room

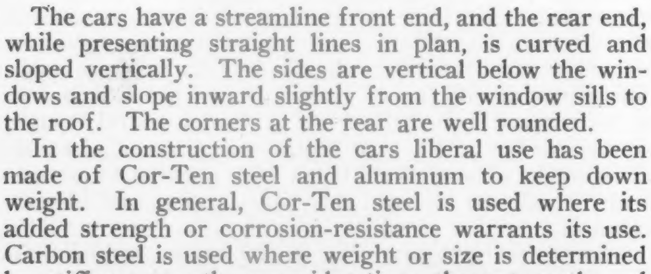
TWO light-weight rail motor cars, designed for single-unit operation, were delivered to the Norfolk Southern by the Brill plant of the American Car & Foundry Company near the end of 1934. These cars, which are known as "rail buses," weigh about 41,000 lb. each and are driven by 176-hp. gas engines through mechanical transmission. They are 56 ft. 7 in. long by 9 ft. wide; 10 ft. 4½ in. in height overall, measured from the rail; provide seats for 53 persons in two passenger compartments, and have a baggage room 12 ft. 11 in. long. The approximate weight distribution is as follows: Body, seats and fittings, 21,500 lb.; power equipment, 5,000 lb.; trucks, 14,900 lb.; passenger and baggage load, 10,000 lb.

These cars were designed after an extended study by the railroad of the requirements for meeting competition by vehicles on highways owned and maintained by the state. They provide a service at low car-mile cost which can be maintained on attractive schedules without excessive first cost.

The Car Structure

The cars have a streamline front end, and the rear end, while presenting straight lines in plan, is curved and sloped vertically. The sides are vertical below the windows and slope inward slightly from the window sills to the roof. The corners at the rear are well rounded.

In the construction of the cars liberal use has been made of Cor-Ten steel and aluminum to keep down weight. In general, Cor-Ten steel is used where its added strength or corrosion-resistance warrants its use. Carbon steel is used where weight or size is determined by stiffness or other considerations than strength and corrosion resistance.



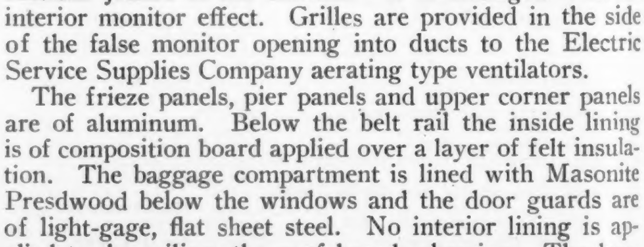
Looking toward the rear end of the car from the forward passenger compartment

The design of the cars for operation as a single unit has obviated the necessity for including a stiff center-sill member. For the underframe and body frame members both rolled and pressed sections are employed as well as some members in truss form. The photographs show clearly how the underframe members have been reduced in weight by cutting away part of the webs of pressed-steel sections. The sides extend below the sill line to within a height of about 12 in. above the rail. The roof is of the plain arch type of wood construction covered with Mulehide roofing and supported by pressed steel carlines. The side sheets and letter panels are of 52S aluminum alloy.

The ceiling is panelled in composition board applied in sections joined at the carlines. It is arranged with an interior monitor effect. Grilles are provided in the side of the false monitor opening into ducts to the Electric Service Supplies Company aerating type ventilators.

The frieze panels, pier panels and upper corner panels are of aluminum. Below the belt rail the inside lining is of composition board applied over a layer of felt insulation. The baggage compartment is lined with Masonite Presdwood below the windows and the door guards are of light-gage, flat sheet steel. No interior lining is applied to the ceiling, the roof boards showing. The baggage compartment is fitted with the customary hardwood chafing strips, secured to the steel lining with Parker-Kalon screws.

All side windows have single lift type sash set in easily removable guides. The sash have metal frames, each adjustable and designed to be rattle-proof and easy to



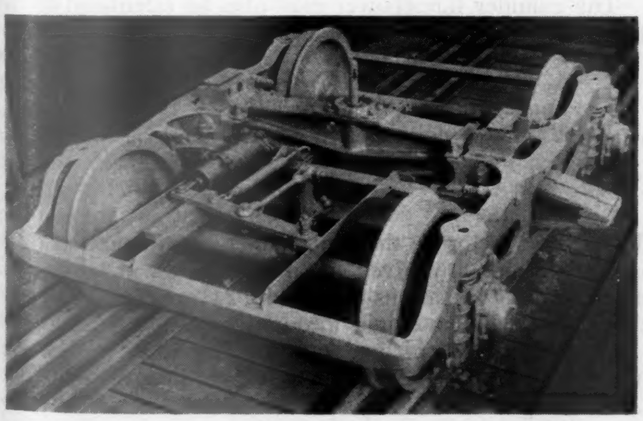
Front ends of two cars showing construction framing

The Norfolk Southern "Rail Bus" built by the American Car & Foundry Company

operate. Six windows are provided around the front end of the car, all except the one on the operator's right having stationary lights set in the body frame. A single drop sash is provided at the operator's side. Two large windows with a vertical dividing rail in the center are provided at the rear of the car. The right-hand window is set stationary in the body frame, while the left window is in two parts, one section adjacent to the center rail being arranged to slide across the other section. The sliding window is provided with a lock designed to hold it dust-tight when closed. The windows directly in front and at the right of the operator are glazed with laminated safety plate glass. All side windows in the passenger compartments are fitted with double faced Pantasote roller curtains.

The passenger entrance doors are located on each side somewhat back of the middle of the car body. The doors are of the automotive parlor-coach type which open outward and, when closed, are flush with the body side sheathing. The floor is 37¼ in. above the rail. A stationary step in a well is provided on each side of the entrance platform with low barriers at the step well which end at the aisle in a vertical stanchion of 1-in. aluminum-alloy pipe. The car is floored with 1⁄16-in. yellow pine throughout. The flooring in the passenger compartments, the platform and the step wells are covered with Armstrong linoleum, an embossed pattern in the aisle and a smooth green tone under the seats. The saloon partitions are of solid sheet steel with hardwood swinging doors. The side doors in the baggage room are flush-mounted, sliding type, each 3 ft. 4 in. wide.

The two passenger compartments are separated by the entrance platform. The one toward the front of the car is for white passengers and seats 30 persons. The rear compartment is for colored passengers and seats 23 persons. The seats in both passenger compartments are Brill No. 309, two-passenger transverse type with individual head rests. A five-passenger transverse seat, similar in construction to the double seats, except that the head rests are omitted, is provided across the rear end of the coach. The seats are upholstered in a machine-buffed leather in a green tone.



Trailer truck of Norfolk Southern "Rail Bus"

There are two saloons, one at the front right-hand corner of the white passenger compartment and one at the front left-hand corner of the colored passenger compartment adjoining the entrance platform. The hoppers are of the Rex dry type. At each saloon is a Giessel water cooler complete with alcove faucet and drain, a Finback, Jr., paper-cup container and a receptacle for used cups.

The lighting is provided by 12-volt circuits. There are nine lamps in the white passenger compartment, seven in the colored passenger compartment, two in the center vestibule and three in the baggage compartment. The dome light fixtures in the passenger compartments have frosted diffusing lenses.

Power Plant and Trucks

The power plant of these cars is of unusual design. It consists of a six-cylinder, Hall-Scott Model 180 en-

(*Continued on page 144*)

View from pit looking up—Hall-Scott engine is mounted entirely underneath the floor

Locomotive Tractive Force in Relation to Speed and Steam Supply[1]

By E. G. Young[2] and C. P. Pei[3]

The authors propose the use of a family of curves to reflect variations in the relation between the four operating variables of speed, cut-off, steam supply and tractive force

THE current interest in the performance and economy of the steam locomotive has focused attention on the old problem of the relation between speed and tractive force and has resulted in some very interesting and important contributions to its clarification.

In most of the methods of estimating tractive force, either empirical or semi-rational, the final relation between the tractive force and speed is represented by a speed-pull curve. A single relation between the tractive force and speed of a given locomotive is an inadequate representation of facts. These facts can only be represented by taking the major variables into account and establishing a series of speed-pull curves on bases that can be rigorously defined. The variables for a given case may be separated into two distinct classes, which, for the present purpose, may be called the "physical factors" and the "operating factors."

When a train is started over a division the work which the locomotive must do at every point is already fixed by the tonnage and the inherent resistances of the train and the speed requirements as known from schedules, train orders, etc., by the profile of the road, and also the weather conditions. The locomotive assigned to the run will presumably be able to meet the requirements imposed upon it, as far as the capacity is concerned at least, and, if the tonnage ratings are well made, with reasonable economy as well. To the physical factors imposed by the train, track and weather may be added the limitations relating to the locomotive itself, its dimensions, proportions and weight. When the crew gets aboard they have under their control only a certain latitude in speed variations, the throttle opening, the cut-off and the firing rate. Eliminating the practice of partial throttle operation because of its inherent inefficiency, there remain only four operating variables; namely, speed, cut-off, firing rate (or steam supply), and the tractive force. Of these four factors fixing any two determines the other two and, consequently, the entire operation of the locomotive, and therein lies the skill of the engine crew. Any two of the four operating factors, such as firing rate and cut-off, may be shown (as in the following paragraphs) to be dependent, if the others are assumed as independent. Hence, the tractive-force-speed relation must be defined by a family of curves, rather than a single relation.

The firing rate and cut-off are not independent factors because the two are related to each other through the total boiler output determined by the firing rate on the one hand, and the total steam requirement for the cylinders determined by the cut-off and speed on the other hand. It, therefore, follows that, at a given speed, the cut-off at which the locomotive may be operated is determined by the available steam supply, which is, of course, determined by the firing rate. Conversely, the firing rate is determined by the steam demand of the cylinders as determined by the cut-off and speed. This leads to the conclusion that the relation between tractive force and speed is specifically defined only when qualified by the operating factors of firing rate or cut-off, or the combination of both.

Since the firing rate and cut-off are the only factors under direct control, either of these may be selected as the starting point in a performance analysis. It is the purpose of this paper to consider the consumption of steam in locomotive cylinders at various speeds and cut-offs, thus determining the steam requirement. No attempt will be made to relate the steam supply to the firing rate, since this relationship rightfully belongs to the study of boiler performance. With the determination of steam requirements and assuming that it can be met by the steam supply from the boiler, it remains to obtain the correlation between the steam supply and the mean effective pressure, from which the cylinder tractive force is calculated. The latter is a physically nonexistent quantity, but it is extremely useful as a basis of comparison. The cylinder tractive force may be defined in relation to the proportion of the locomotive by the familiar expression

$$T = p \cdot \frac{d^2 s}{D} \quad \text{.......} \quad (1)$$

where

T = cylinder tractive force in lb.
p = mean effective pressure in lb. per sq. in.
d = diameter of cylinder in in.
s = length of stroke in in.
D = diameter of driving wheels in in.

The quantity d^2s/D is constant for any given locomotive and will be designated as Z.

The cylinder horsepower may also be calculated from the mean effective pressure as follows:

$$h = p \cdot \frac{4\,a\,s}{12 \times 33{,}000} \cdot N \quad \text{.......} \quad (2)$$

where

h = cylinder horsepower
a = area of piston in sq. in.
N = speed in r.p.m.

The second factor on the right-hand side of equation (2) is constant for any given locomotive and will be designated as Y. Equations (1) and (2) apply to conventional two-cylinder single locomotives.

Many methods of making tractive force estimates have been proposed which depend on the relation

$$h = \frac{E}{W} \quad \text{.......} \quad (3)$$

[1] Abstract of a paper presented before the 1934 annual meeting of the American Society of Mechanical Engineers.
[2] Research Professor, Railway Mechanical Engineering, University of Illinois, Urbana, Ill.
[3] Champaign, Ill.

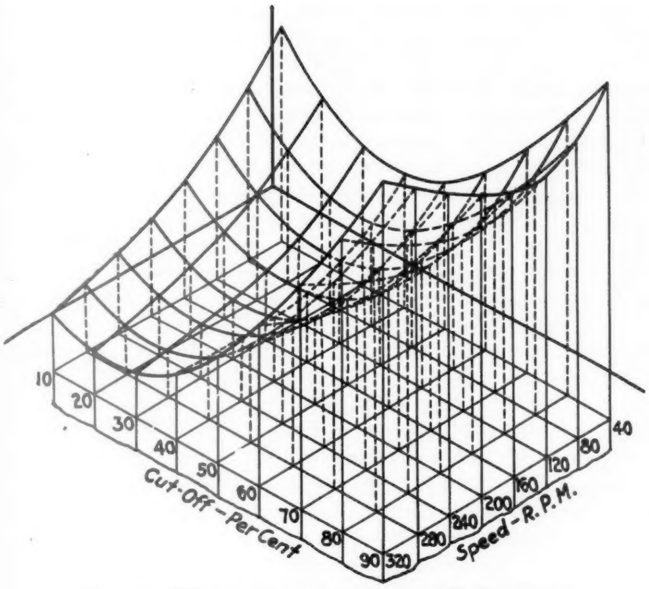


Fig. 1—Water-rate—speed—cutoff diagram

where

E = hourly evaporation in lb.
W = water rate in lb. per I. hp.

This is, of course, a true relation, but in view of the difficulty of estimating W the process is extremely indirect. The fact is well known that at starting, with low speed and maximum cut-off, the water rate is high, of the order of 30 lb., also that at high speed and short cut-off minimum values are obtained, of the order of 15 lb. or lower. It is evident that in speeding up a train the water rate passes through this entire range and several investigations have correlated water rate with speed. This process, however, is inadequate since at the same speed operation at a considerable range of cut-offs is possible, depending on the steam supply; and the cut-off is also a factor in determining water rate, of equal importance with the speed. Sufficient data are now available so that a representative diagram for water rates, for locomotives working at about 200 lb., boiler pressure can be drawn. When it is drawn as a surface relating the water rate in pounds per indicated horsepower, speed in revolutions per minute, and cut-off in per cent of stroke, its form is shown in Fig. 1. Water-rate curves are usually shown plotted against the speed, as represented by the light lines in Fig. 2. The group of points through which various lines of equal cut-offs are drawn in the figure have been in some investigations merely enclosed by an area of steam performance, and in other cases a still less valid representation has been made in which all values for a given speed are averaged and the average points connected. It is obvious that the average relation will depend more on the number of varying speed-cut-off combinations than upon the specific results.

If a relation between speed and water rate, independent of cut-off, is assumed, such a relation will cut across the curve in Fig. 2, as shown by the heavy line XY. Many such relations have been assumed, each correct in itself provided the locomotive is operated through the program of speed-cut-off combinations upon which the water-rate relation is based. But the forecasting of such a program as a generalization can only be a matter of opinion, and the actual results in any given case will be closely related to the method of operation governed by the variations in operating conditions.

If any fixed curve XY in Fig. 2 is to be discounted as an unsatisfactory generalization, the scheme of computing the tractive force from the water rate must also be rejected as the general relation, as shown by the light line, cannot be used directly. For example, it is desired to calculate the tractive force of a locomotive of known proportions at a given speed; let the conditions be defined by assuming a cut-off. From Fig. 2 a water rate may be assumed accordingly, but an additional process must be provided before the horsepower or tractive force can be determined. Similarly, if a steam supply is assumed, the steam available per revolution is determined, but, without the additional process, horsepower still remains unknown.

The proposed method is a more direct one, by means of which may be calculated:

a—The total amount of steam used per hour when the cylinder dimensions, working pressure, speed and cut-off are known or assumed.

b—The cut-off at which the locomotive may be operated when the pressure, cylinder dimensions, speed, and steam supply are known.

c—The relation between speed, cut-off and mean effective pressure as a fraction of the admission pressure, so stated that, with any two quantities given, the third one may be determined.

The process of estimating the tractive force as applied to a locomotive of known dimensions would be as follows:

1—Assume the speed and cut-off.
2—Determine the steam used per revolution from (a) above.
3—Determine the steam required per hour.
4—Determine the admission pressure, by finding the probable loss between the boiler and cylinders.
5—Find the mean effective pressure from (4) and (c) above.

If, in place of assuming the cut-off, the steam supply is assumed, the process becomes:

1—Assume speed and steam supply per hour.
2—Find the steam per revolution from the hourly supply and speed.
3—Find the cut-off which may be used with this amount of steam per revolution from (b) above.
4—Determine the admission pressure, as before.
5—Find the mean effective pressure, as before.

The development of the process depends entirely on specific information obtained from the laboratory test

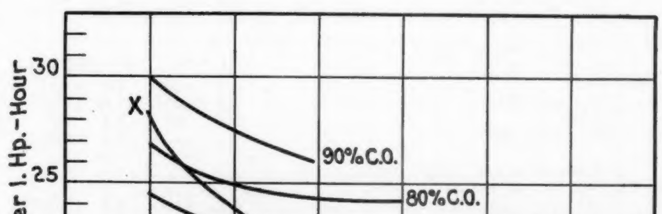


Fig. 2—A family of water-rate and speed curves

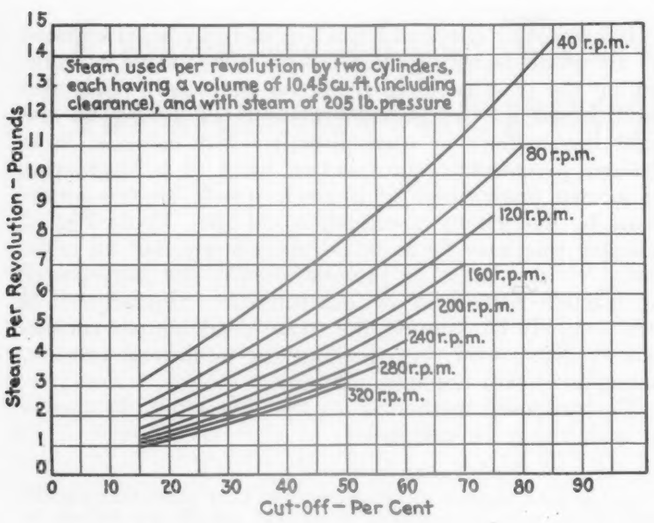


Fig. 3—Base steam-consumption curves

results, eliminating the uncertainties resulting from average steam output, average speed, average tractive force and average water rate usually obtained in road tests. The laboratory data are by no means as complete as might be desired, and in many cases inconsistencies make the determination of certain quantities most difficult. However, enough data are available in published form to make useful generalization possible, and it is hoped that the remaining gaps may be filled by additional tests.

The plot in Fig. 3 shows the relation between the steam per revolution, the cut-off and the speed in r.p.m. for the two Pennsylvania locomotives, class L1s[4] and lap in the tests at 120 and 160 r.p.m. only, but the results for these speeds are so consistent as to make it legitimate to consider the combined results as coming from a single locomotive, tested over a wider range of speed and cut-off conditions than was ever used in a series of locomotive laboratory tests. These locomotives have cylinder volumes, including clearance, of 10.7 and 10.2 cu. ft. (one cylinder), respectively. No different relations may be drawn at the overlapping speeds for the two locomotives and, hence, the volume of 10.45 cu. ft. may be taken as representative of the two. It remains to determine what relation is borne to the steam used in the 10.45-cu.-ft. cylinders and at 206-lb. boiler pressure by that used in cylinders of other volumes and with steam at other pressures.

Test reports are also available for a series of locomotives[6] using superheated steam and with cylinder volumes from 6.5 cu. ft. to 10.7 cu. ft., all working at the same boiler pressure of 205 lb. per sq. in.; of a locomotive with three 8-cu.ft. cylinders and 200 lb. boiler pressure[7]; of two locomotives with 14.6-cu.ft. cylinder volumes and 250 lb. boiler pressure[8]; and of the Baldwin 60,000[9] with one high-pressure cylinder of 11.7 cu. ft. volume, with boiler pressure of 250 and 350 lb. per sq. in. Some data are also available for the performance of the Purdue No. 4, with 3.5-cu.ft. cylinder volume. All volumes are those of one cylinder, including clearance. Curves similar to those of Fig. 3 were drawn throughout the range of performance for all of these locomotives; these curves are not presented, but in Table I are shown the values from smooth curves drawn through the actual points and also the percentage which the steam used represents of that used by the L-K combination at the same speed and cut-off. From this table the following summary may be condensed:

[4] Pennsylvania Railroad Bulletin No. 28.

[5] Pennsylvania Railroad Bulletin No. 29.

[6] Pennsylvania Railroad locomotives E3sd, E6s-89, E6s-51, K2sa, and H8sb, see Pennsylvania Railroad Bulletins 11, 21, 27, 18 and 10, respectively.

[7] Missouri Pacific locomotive No. 1690, reported in Railway Age, June 30, 1925.

[8] Pennsylvania Railroad locomotives I1s-790 and I1s-4358, see Pennsylvania Railroad Bulletins 31 and 32, respectively. These two locomotives differed in that the latter one had a Type E superheater and also a feed-water heater, but the resulting difference in performance does not enter into consideration of this paper.

[9] Baldwin Locomotive Works publication.

Table I—Steam Per Revolution for Various Locomotives

		Cut-off, Percent															
		20		30		40		50		60		70		80		90	
R.P.M.	Locomotive (Cyl. Vol., Pressure)	Lb.	Ratio	Lb.	Ratio	Lb.	Ratio	Lb.	Ratio	Lb.	Ratio	Lb.	Ratio	Lb.	Ratio	Lb.	Ratio
40	I1s (14.6, 250)	6.6	1.69	8.6	1.65	10.7	1.62	13.0	1.60	15.6	1.61	..	...	..	...	..	...
40	L1s and K4s (10.45, 205)	3.9	1.00	5.2	1.00	6.6	1.00	8.1	1.00	9.7	1.00	..	...	..	...	15.8	1.00
40	H8sb (8.5, 205)	3.4	.87	4.7	.90	5.2	.79	6.4	.79	7.6	.78	..	...	..	...	9.3	.59
40	Mo. Pac. 1690 (8.5, 200)	..	...	..	...	...	...	..	...	..	...	..	...	..	...	12.0	.76
80	Baldwin 60,000 (11.7, 350)	..	...	..	...	...	...	8.5	1.37	10.8	1.44	13.4	...	..	...	20.2	1.68
80	Baldwin 60,000 (11.7, 250)	..	...	..	...	...	...	..	...	..	...	9.2	...	..	...	14.0	1.17
80	I1s	4.3	1.59	6.6	1.58	7.6	1.52	9.5	1.53	..	...	..	...	..	...	...	...
80	L1s and K4s	2.7	1.00	3.8	1.00	5.0	1.00	6.2	1.00	7.5	1.00	..	...	..	...	12.0	1.00
80	H8sb	2.4	.89	3.4	.89	4.5	.90	5.5	.89	6.6	.88	..	...	..	...	...	...
80	Mo. Pac. 1690	..	...	3.1	.82	3.9	.78	..	...	..	...	..	...	..	...	...	...
120	Baldwin 60,000 (11.7, 350)	..	...	..	...	...	...	6.9	1.30	8.8	1.35	11.0	...	13.6	...	17.2	...
120	Baldwin 60,000 (11.7, 250)	..	...	..	...	...	...	..	...	..	...	..	...	8.8	...	...	...
120	I1s	3.5	1.52	4.8	1.50	6.2	1.48	7.9	1.49	9.7	1.49	..	...	..	...	...	...
120	L1s and K4s	2.3	1.00	3.2	1.00	4.2	1.00	5.3	1.00	6.5	1.00	..	...	..	...	...	...
120	H8sb	2.0	.87	2.7	.84	3.7	.88	4.5	.85	..	...	..	...	..	...	...	...
120	Mo. Pac. 1690	..	...	..	...	...	...	..	...	4.8	.74	..	...	..	...	...	...
120	E6s-89 and E3sd (6.5, 205)	..	...	..	...	...	...	..	...	..	...	..	...	..	...	...	...
120	Purdue No. 4 (3.5, 170)	..	...	1.5	.47	...	...	2.0	.38	...	...	..	...	..	...	...	...
160	Baldwin 60,000 (11.1, 350)	..	...	..	...	...	...	5.8	1.26	7.4	1.30	9.4	...	..	...	...	...
160	Baldwin 60,000 (11.1, 250)	..	...	..	...	...	...	..	...	..	...	5.8	...	..	...	...	...
160	I1s	..	...	3.75	1.34	5.0	1.39	6.6	1.44	..	...	..	...	..	...	...	...
160	L1s and K4s	..	...	2.8	1.00	3.6	1.00	4.6	1.00	5.7	1.00	..	...	..	...	...	...
160	H8sb	..	...	2.4	.86	3.2	.89	4.0	.87	..	...	..	...	..	...	...	...
160	Mo. Pac. 1690	..	...	2.2	.79	2.8	.78	3.4	.74	4.2	.74	..	...	..	...	...	...
160	E6s-89 and E3sd	..	...	2.2	.79	2.9	.81	3.7	.81	..	...	..	...	..	...	...	...
160	Purdue No. 4	..	...	..	...	...	...	..	...	..	...	..	...	..	...	...	...
200	K4s	1.7	1.00	2.5	1.00	3.4	1.00	4.2	1.00	5.4	1.00	..	...	..	...	...	...
200	E6s-89	1.4	.82	2.0	.80	2.7	.79	3.5	.83	4.4	.82	..	...	..	...	...	...

Locomotive	Cylinder volume: Cu. ft.	Cylinder volume: L-K	Working pressure: Lb. per sq. in.	Working pressure: Ratio to L-K	Conditions of comparison	Steam, ratio to L-K
P.R.R. E3sd and E6s-89	6.5	.63	205	1.00	All speeds, all cut-offs	.80
Mo. Pac. 1690	8.0	.75	190[10]	.93	All speeds, all cut-offs	.77
P.R.R. H8sb	8.8	.82	205	1.00	All speeds, all cut-offs	.87
Baldwin 60000	11.7	1.10	335[10]	1.65	All speeds,	1.50
			200[10]	.95	all cut-offs	1.06
P.R.R. I1s	14.6	1.36	250	1.21	All cut-offs, speeds:	
					40 r.p.m.	1.60
					80 r.p.m.	1.50
					120 r.p.m.	1.50
					160 r.p.m.	1.40
Purdue No. 4	3.5	.33	170	.83	Speeds, 120 and 160 r.p.m. Cut-offs, 30 and 40 per cent	.45

From the foregoing tabulation Fig. 4 is plotted. The circles represent the various locomotives, showing for each its average ratio of steam used to that for the L-K combination, plotted against the volume ratio. The points representing locomotives with 205 lb. working pressure show the use of less steam than that for the L-K combination in a ratio related to the smaller cylinder volume. The points representing locomotives with other than 205 lb. pressure show a use of more or less steam dependent both on the cylinder volume and working pressure. The first step in relating these is to reduce the varying pressure points to the values they would have if 205 lb. pressure were used. By trial it was found that when the values of these points were divided by the quantity

$$0.2 + 0.8 \frac{P_2}{P_1}$$

where P_2 is the actual working pressure of the locomotive under consideration and P_1 is the working pressure of the L-K combination, or 205 lb., the corrected points fall close to the straight line drawn through the 205-lb. pressure points. The equation for the latter line shows that, if the steam for the L-K combination with the cylinder volume V_1 is taken as unity, the steam for a locomotive with some other cylinder volume V_2 is

$$0.3 + 0.7 \frac{V_2}{V_1}$$

Hence, the steam per revolution S_2 for a locomotive with pressure P_2 and volume V_2 as compared with the steam S_1 used by the L-K combination with pressure P_1 and volume V_1 may be expressed as follows:

$$\frac{S_2}{S_1} = \left(0.3 + 0.7 \frac{V_2}{V_1}\right) \left(0.2 + 0.8 \frac{P_2}{P_1}\right) \qquad (4)$$

This is a purely empirical relation based on test results. The variations in the ratios for any given locomotive within its own range of performance are not sufficiently consistent for any other process to define this relation and, in general, the variation secured by using the formula proposed for finding S, the steam used per revolution, is less than the variation in the test data.

The value of S thus secured represents the steam used per revolution for both cylinders of the conventional two-cylinder simple locomotives; for three-cylinder simple locomotives, such as the Missouri Pacific No. 1690, the steam used per revolution is 50 per cent greater than S, or 1.5 S, and for locomotives with only one high-pressure cylinder, as in the case of Baldwin 60000, the steam used per revolution is equal to 0.5 S.

After determining the amount of steam per revolution it becomes necessary to find the admission pressure in order that the mean effective pressure may be later determined. From the preceding process S ($= S_2$) is known, and the total steam used by the cylinders in pounds per hour is

$$E = 60\ N\ S \qquad (5)$$

where

E_c = steam used by the cylinders in lb. per hr.
N = speed in r.p.m.
S = steam used by the cylinders per revolution, in lb.

The pressure of the steam as delivered to the cylinders varies with the steam flow and with the area of all of the passages from the boiler to the cylinders. These areas are at least four in number and there are no data or theory to evaluate their effect. However, the loss in pressure, for any given steam supply, is closely related to the capacity of the boiler and this, in turn, to its dimensions, so that a satisfactory correlation is found between the loss in pressure, the steam flow and the heating surfaces. This relation is well represented by the expression

[10] In calculating the ratios attention was paid to the actual range of boiler pressures in the tests. On this basis the working pressure for the Missouri Pacific locomotive No. 1690 is 190 lb., and for the Baldwin 60000, 335 and 200 lb.

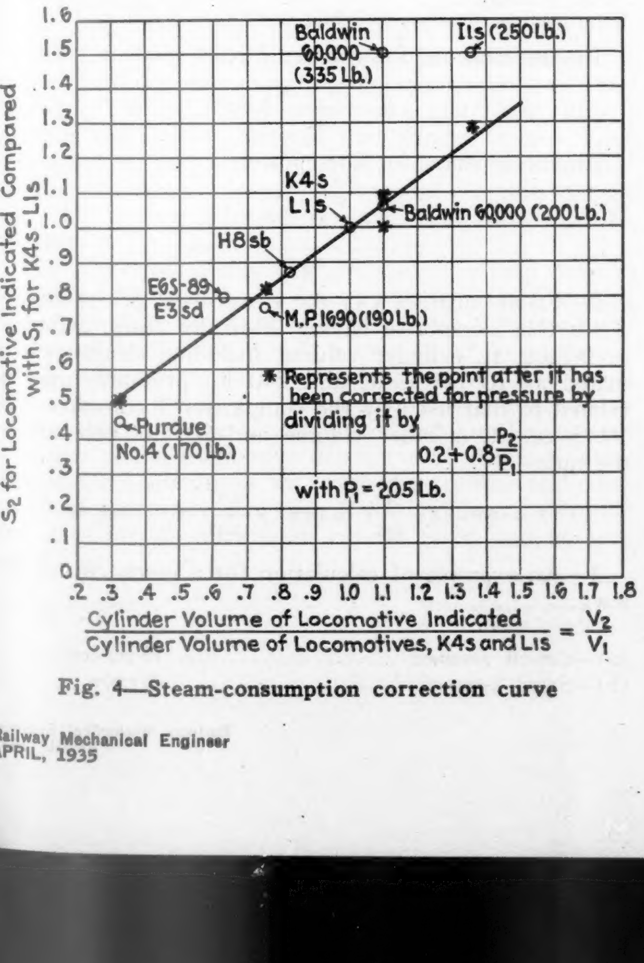


Fig. 4—Steam-consumption correction curve

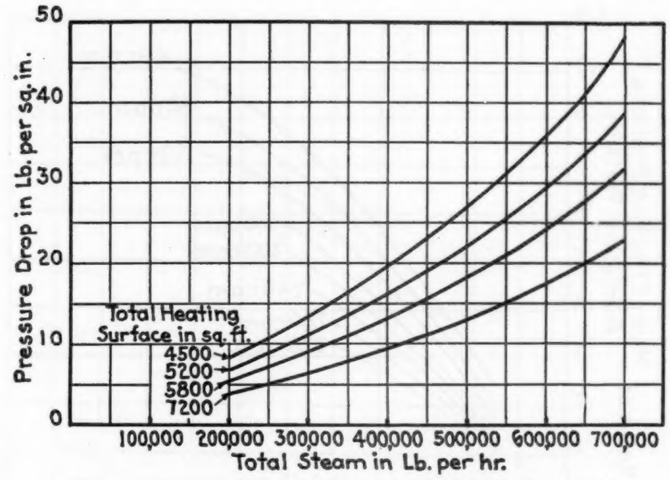


Fig. 5—Steam-pressure drop between boiler and branch pipe

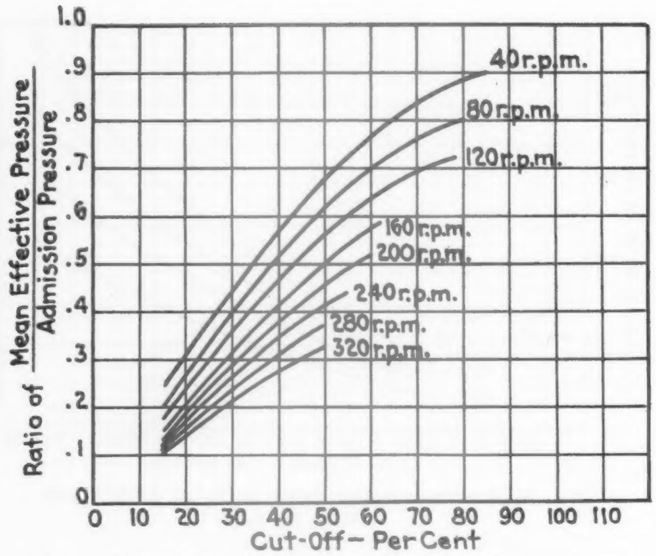


Fig. 6—The effect of cutoff on mean effective pressure

$$P_d = 0.6 \left(\frac{E}{H}\right)^{1.6} \quad \text{.......} \quad (6)$$

where

P_d = loss in pressure between the boiler and branch pipe, in lb. per sq. in.
E = total steam consumption in lb. per hr.
H = total heating surfaces (outside), including that of superheater, in sq. ft.

It is quite obvious that the total steam consumption E is made up of the steam used in the cylinders E_c and the steam used by the auxiliary devices, E_a, or

$$E = E_c + E_a$$

The curves from which equation (6) is derived are shown in Fig. 5. They are valid only up to the point at which the boiler pressure itself begins to fall off.

For any simple locomotive operating at a given speed the work in the cylinders depends on the mean effective pressure, which in turn depends on the admission pressure, the cut-off and the steam distribution generally. No attempt can be made to estimate the effects of the variations in steam distribution design, and the arrangements of the conventional engines are such that this variable may be eliminated. The effect of the "diagram factor" is so consistent that the mean effective pressure is, for any given speed and cut-off, a very definite proportion of the admission pressure. The ratios of the mean effective pressure to the admission pressure at the various cut-offs were plotted for the P. R. R. locomotive L1s, K4s, H8sb and I1s, I. C. locomotive No. 1742[11], and Missouri Pacific locomotive No. 1690 for the usual range of tests speeds. The curves thus obtained accurately represent the range and value of the ratios. In the case of the four lower speeds there are several curves available for comparison; for the higher speeds the information from K4s tests is used. These curves are assembled in a composite diagram in Fig. 6, which may be taken as representing the relation between the ratio of mean effective pressure to admission pressure, and the cut-off in per cent of stroke, at the various speeds in revolutions per minute, for locomotives with conventional types of cylinders and valve gears and using superheated steam.

The process of estimating the tractive force and horsepower of a locomotive may now be summarized:

[11] University of Illinois, Engineering Experiment Station Bulletin No. 220.

Case (A)—Given the dimensions of the locomotive, the speed and the cut-off, the procedure is as follows:

(a)—From Fig. 3 find the steam per revolution for the locomotive used as standard of comparison, having 10.45 cu.ft. cylinders and 205 lb. working pressure, at the required speed and cut-off.

(b)—Find the steam per revolution for the pressure and cylinder volume of the locomotive considered by modifying the steam per revolution found under (a) by the use of equation (4).

(c)—Find the steam used by the cylinders per hour by the use of equation (5).

(d)—Find the total steam used per hour by adding an estimated amount of steam used by the auxiliary devices.

(e)—From the total steam demand and the heating surfaces estimate the pressure drop between boiler and cylinders by the use of equation (6) or Fig. 5.

(f)—Find the admission pressure by deducting the pressure drop (e) from the boiler pressure.

(g)—Estimate the ratio of mean effective pressure to admission pressure for the given speed and cut-off from Fig. 6 and admission pressure (f) by this ratio.

(h)—Calculate the tractive force from equation (1).

(i)—Calculate the cylinder horsepower from equation (2).

Case (B)—Given the steam supply and the speed the procedure is as follows:

(a)—If the steam supply is given as the net amount to the cylinders, the steam per revolution is obtained directly by dividing the steam supply E_c by 60 N, otherwise the total steam supply must be reduced by an estimated amount of steam used by the auxiliary devices in order to arrive at the net amount of steam to the cylinders.

(b)—Reduce the actual steam per revolution (a) by the use of equation (4) to find the amount of steam per revolution required by the locomotive used as the basis of comparison.

(c)—Knowing the speed and the steam per revolution (b) find the cut-off at which the locomotive may be operated for the speed from Fig. 3.

Further procedure is the same as the items (d) to (i), both inclusive, previously outlined under case (A), since the speed, cut-off and total steam supply are now known.

As a means of showing the application of the method just outlined, the cylinder tractive force and cylinder horsepower of a locomotive of assumed dimensions will be determined for a series of assumed values of cut-off and steam supply.

1—Locomotive dimensions:

Cylinders	2, 26 in. by 28 in.
Driving wheel, diameter of	80 in.
Boiler pressure	240 lb. per sq. in.
Total heating surfaces, including superheater	6,000 sq. ft.

Engine constant for tractive force:

$$Z = \frac{26^2 \times 28}{80} = 237$$

Engine constant for horsepower

$$Y = \frac{4 \times \frac{\pi}{4} 26^2 \times 28}{12 \times 33{,}000} = 0.150$$

2—From equation (4) the steam used per revolution S for this locomotive having 26-in. by 28-in. cylinders, or 9.5 cu. ft. cylinder volume, including clearance estimated at 10 per cent, and 240 lb. pressure, may be related to that used by the comparison locomotive with 10.45 cu. ft. cylinder volume and 205 lb. pressure by the ratio

$$S = \left(0.2 + 0.8 \times \frac{240}{205}\right) \left(0.3 + 0.7 \times \frac{9.5}{10.45}\right) S_1 = 1.064 \ S_1$$

3—An example of calculation for a given cut-off follows:

(a)—Cut-off assumed	75 per cent
(b)—Speed assumed	40 r.p.m.

(c)—Steam per revolution for the comparison locomotive with 10.45 cu.ft. cylinder volume and 205 lb. pressure, from Fig. 3 ... 12.4 lb.
(d)—Steam per revolution for the locomotive with 9.5 cu.ft. cylinder volume and 240 lb. pressure, (c) × 1.064.... 13.2 lb.
(e)—Steam to cylinders per hour, (d) × (b) × 60 .... 31,700 lb.
(f)—Estimated total steam used, including auxiliary devices .... 35,000 lb.
(g)—Pressure drop between boiler and branch pipes, calculated from equation (6), with E = 35,000 and H = 6,000.. 10 lb. per sq. in.
(h)—Admission pressure, 240 — (g)........ 230 lb. per sq. in.
(i)—Ratio of mean effective pressure to admission pressure, for 40 r.p.m. and 75 per cent cut-off, from Fig 6............ 0.86
(j)—Mean effective pressure (h) × (i).... 198 lb. per sq. in.
(k)—Cylinder tractive force, (j) × 237.. 47,000 lb.
(l)—Cylinder horsepower, (j) × 0.150 × (b) 1,188
(m)—Water rate, lb. per i.hp.hr., (e) ÷ (l). 26.7 lb.

The calculations of cylinder horsepower and water rate, in addition to the tractive force, are carried out as a check on the validity of the result.

4—A similar set of calculations based on steam supply may be made, thus:

(a)—Steam to cylinders, assumed.......... 40,000 lb. per hr.
(b)—Total steam required, including the estimated steam for auxiliary devices.. 44,000 lb. per hr.
(c)—Speed, assumed .... 240 r.p.m.
(d)—Steam per revolution for this locomotive, (a) ÷ 60 (c)................ 2.78 lb.
(e)—Steam per revolution based on the comparison locomotive, (d) ÷ 1.064.. 2.61 lb.
(f)—Cut-off corresponding to 240 r.p.m. and 2.61 lb. steam per revolution, from Fig. 2 .... 38 per cent
(g)—Pressure drop, from equation (6).... 14 lb. per sq. in.
(h)—Admission pressure, 240 — (g)....... 226 lb. per sq. in.
(i)—Mean effective pressure ratio corresponding to 240 r.p.m. and 38 per cent cut-off, from Fig. 6 .... 0.328
(j)—Mean effective pressure, (h) × (i).... 74 lb. per sq. in.
(k)—Cylinder tractive force, (j) × 237.... 17,500 lb.
(l)—Cylinder horsepower, (j) × 0.150 × (c) 2,560
(m)—Water rate, lb. per i.hp.hr., (a) ÷ (l) 15.6

Similar calculations have been carried out with the cut-offs assumed at 75, 60, 45, and 30 per cent, and with steam to the cylinders assumed at 30,000, 40,000, 50,000 and 60,000 lb. per hr. For every assumed cut-off or steam supply the calculations are made for the speeds of 40, 80, 120, 160, 200, 240, and 280 r.p.m., omitting some of the obviously impracticable combinations of speed and cut-off. The calculated results are plotted in Fig. 7 showing the relation between the cylinder tractive force and speed in two sets of curves; one for the tractive-effort-speed relation at various constant cut-offs, and the other for various rates of constant steam supply to the cylinders.

The proposed methods of calculation extend only to a speed of 40 r.p.m. The determination of tractive force at low speeds rests on the fact that for the first few revolutions of the driving wheels a very small steam supply will be adequate to fill the cylinders, so for zero speed when, so to speak, the locomotive is merely leaning against the train, the actual cut-off and the dimensional constants of the locomotive determine the tractive force. Reasonable assumptions with regard to release and compression pressures give the following mean effective pressures for the maximum cut-off given and at zero speed.

Maximum cut-off, per cent	Mean effective pressure ÷ boiler pressure
90	.97
80	.93
70	.87
60	.80

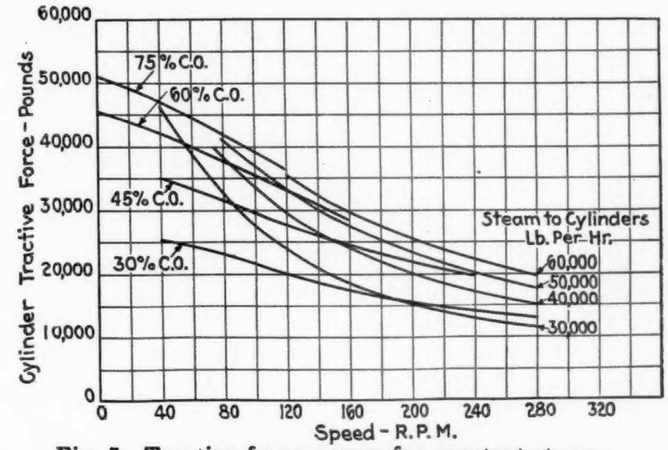


Fig. 7—Tractive-force curves for constant-steam-supply and constant-cutoff

The advantages of the method proposed and of the resulting figure may be mentioned briefly as follows:

(a)—The process takes the physical facts of the transformation from potential into kinetic energy fully into account.

(b)—The resulting chart places before the user a full exhibit of the capabilities of the locomotive.

(c)—If such a chart is to be used as the background for a single-line speed-pull curve, it is immediately apparent what program of evaporative capacity and cut-off is expected of the locomotive.

(d)—A basis is furnished for estimating the economy of the locomotive under any conditions, including those far inside of what may be considered its capacity.

4-8-4 Heavy Fast Locomotives For the Lehigh Valley

(Continued from page 135)

Franklin lateral-motion driving boxes are applied on the second pair of drivers in order to reduce the rigid wheel base and permit of passing through sharp curves. Wedges are of the Franklin adjustable type. Driving axles are hollow bored.

On the first three locomotives Timken roller bearings were provided on the front truck axles only. On the last two locomotives Timken roller bearings were fitted to all axles—front truck, driving axles, trailing truck and tender trucks. Driving-box safety bombs are provided.

Driving Gear

The cylinders are 27 in. in diameter with 30 in. stroke. The valve gear is of the Baker long-travel type and provides movement to 12-in. piston valves having a maximum travel of 8⅝ in. and a cut-off in full gear of 82½ per cent. A Franklin power reverse gear is mounted on a bracket which is a part of the bed casting and is thus carried free from the boiler. Control is by means of a Bradford front-end throttle valve furnished by the American Throttle Company. Lunkenheimer drifting valves and Walworth cylinder relief valves are provided. The crossheads are of the Laird multiple-ledge type. Piston-rod and valve-stem packings are of the U. S. Metallic Packing Company King type.

The brakes are of the new No. 8-ET type furnished by the Westinghouse Air Brake Company. The locomotives are also equipped with automatic train control furnished by the General Railway Signal Company.

A Detroit mechanical lubricator is provided with feed pipes to the cylinders, valves, guides, stoker, driving-wheel hubs, trailing-wheel hubs, throttle valve and drift-

ing-valve rigging, headlight generator and—by means of a special hose—to the stoker trough on the tender. A Bosch type lubricator is applied for the air compressors. Fittings are also provided for pressure grease lubrication of many parts.

The rectangular type tenders, which are mounted on water-bottom cast-steel underframes furnished by the General Steel Castings Corporation, have a capacity for 20,000 gallons of water and 30 tons of soft coal. They are carried on six-wheel trucks furnished by the General Steel Castings Corporation which are fitted with clasp brakes furnished by the American Steel Foundries.

The general dimensions and weights of these locomotives are given in one of the accompanying tables. The weights given are for the first three locomotives. On the last two locomotives equipped with roller bearings throughout the weight of the engine is slightly greater, the weight on the drivers being a little less and that on the trucks a little more.

Light-Weight Motor Cars On the Norfolk Southern

(*Continued from page 137*)

gine, with 5-in. by 6-in. horizontal cylinders, which develops a maximum of 176 hp. at 1,800 r.p.m. This engine is mounted entirely beneath the floor; in fact, no space is required above the floor for any of the mechanical equipment of the car, except for the controls adjacent to the operator's station. The carburetor is a Zenith balanced type, attached to a hot-spot manifold, and has a large oil-wetted type air cleaner. The starting motor is a 12-volt Bendix type. The generator is a 12-volt, 1,000-watt Delco-Remy unit. The engine drives through a Brown-Lipe three-speed, helical-gear, constant-mesh transmission and a Long double-plate clutch 14 in. in diameter to a single driving axle at the forward end of the front truck. A drive shaft and a propeller shaft lead from the transmission to the drive axle. The drive shaft is coupled to the transmission by a fabric disk joint and is supported at the rear end by a self-aligning ball bearing. The propeller shaft is provided with two needle type, universal and slip joints.

The driving axle is provided with a gear case in which are two free running bevel gears meshing with a common pinion. One or the other of the bevel gears is secured against rotation of the axle by shifting a large clutch operating on a splined center portion of the axle, thereby selecting either forward or reverse movement. The operator controls the car from his seat on the right side at the front of the baggage room, through a foot-operated clutch and hand-operated gear shift. The spark control is automatic and the throttle is controlled by foot-accelerator and auxiliary hand control.

Fuel is supplied to the carburetor by a pressure pump from a 100-gallon heavy gage tank under the floor.

A fin-tube radiator is mounted under the car floor beneath the center of the car and behind the engine. The major portion is cooled by draft created by a fan and the remainder by natural draft. The four-blade propeller-type fan is driven from the engine. The car is heated through light fin-tubing radiators along the sides of the passenger compartment, through which water from the engine jacket is circulated.

The trucks are both of Brill design with 6-ft. wheel base and are carried on 30-in. One-Wear rolled-steel wheels. The axle for the driving wheels is heat-treated alloy steel of special design to suit the driving mechanism. All the axles have Timken roller-bearing journals.

The trucks have cast-steel side frames with integral pedestals and cast-steel bolsters. The power truck is arranged with the bolster off center so that the driving axle carries about 59 per cent of the center-plate load and 34 per cent of the total car weight. A coil spring is inserted in each pedestal over the journal box. The journal boxes are guided in the pedestal by shear-type rubber blocks, one on each side of the box which, under normal operation, transmit all forces between the axle and the truck frame. In case of excessive lateral forces or high braking pressures, however, the rubber is deflected sufficiently to bring phosphor-bronze pedestal liners into play. Rubber inserts are also provided between the end clips of the elliptic bolster springs to insulate the top and bottom sections from each other.

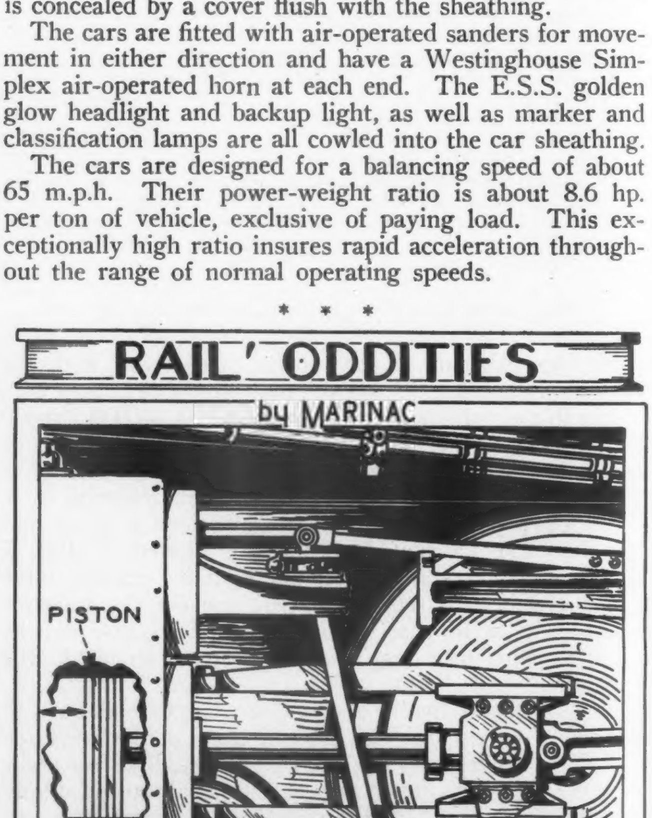
The cars are equipped with Westinghouse SME type air brakes using a self-lapping brake valve arranged with both hand- and foot-operated safety control. Air is furnished by a 10 cu. ft. high-speed compressor driven directly from the engine shaft. The brake rigging is of the outside-hung type on the driving truck with the brake cylinder mounted on the car body. On the trailer truck the brakes are inside-hung with the cylinder mounted on the truck.

An A.R.A. standard pocket coupler is provided at each end of the car. The coupler heads are removable and intended for use in emergency only. Each coupler pocket is concealed by a cover flush with the sheathing.

The cars are fitted with air-operated sanders for movement in either direction and have a Westinghouse Simplex air-operated horn at each end. The E.S.S. golden glow headlight and backup light, as well as marker and classification lamps are all cowled into the car sheathing.

The cars are designed for a balancing speed of about 65 m.p.h. Their power-weight ratio is about 8.6 hp. per ton of vehicle, exclusive of paying load. This exceptionally high ratio insures rapid acceleration throughout the range of normal operating speeds.

* * *



Further information furnished by the editor upon request

EDITORIALS

What Will Become of the Mechanical Associations?

The old Master Car Builders' Association and the American Railway Master Mechanics' Association, now combined and known as the Mechanical Division of the Association of American Railroads, have concrete, constructive accomplishments to their credit, concerning which the members of the two original associations can well be proud. We must not forget that these associations were not started by the railroads, but were inaugurated and controlled by individual mechanical-department officers, who recognized the need of working intimately together in improving the mechanical equipment and facilities of the railroads, and of making the cars available for interchange. It would be difficult to overvalue the work of these two organizations in building the American railway system to its present high standards.

The membership of the two associations overlapped to a considerable extent and for this reason they co-operated intimately in many respects, and particularly in the holding of their annual conventions. Eventually, and possibly wisely, they were forced to unite into one organization, and became a part of the American Railway Association. Individual memberships were replaced by railroad memberships, and under war conditions, and later under economic stress, annual conventions of the association were either canceled or were replaced by annual meetings of the General Committee.

There can be no question but what under such conditions something was lost in the way of morale and individual aggressiveness. The individual railroad mechanical officer became a cog in the machine, although there is some contention that a greater degree of independence and aggressiveness might have been retained. The engineering officers have maintained the American Railway Engineering Association intact and have continued to hold annual meetings, even though these may have been curtailed in length. Surely, under present conditions, when the mechanical department has so many difficult problems confronting it, there is every bit as much reason why its association should continue to function, as for the engineering association to do so.

The old Master Car Builders' and Master Mechanics' Associations included in their programs only the more important problems in these two departments, and did not pretend to get down to matters of shop practice, or such things as locomotive operation and car inspection. It was not surprising, therefore, that over the years, ambitious and energetic minor mechanical-department officers followed the lead of the heads of the department and formed a number of organizations which held annual conventions, which were usually accompanied by small exhibits. The reports and proceedings of these meetings had a wide circulation and were highly regarded for their practical suggestions and value. Even more than this, the opportunity of the officers and foremen to get together to discuss their common problems, proved to be a real inspiration, because of friendships which were engendered and the informal interchange of ideas and experiences, not always included in the formal record.

Naturally, when, under stress, the major associations slowed down their convention programs, these minor mechanical associations suffered even more severely. Some of them have not held meetings for several years and their memberships have been hard hit because of changes which have taken place in personnel. At the meeting of the General Committee of the Mechanical Division last June, its members felt so strongly about the value of conventions of these organizations, that it was decided to encourage them at least partly to resume their former activities. In September, representatives of the various associations met in Chicago and decided tentatively to hold conventions early in May of this year, four of the associations meeting individually and simultaneously on the last three days of one week, and four on the first three days of the following week. It was decided, also, by the allied railway supply groups to have a common exhibit for all of the meetings.

Railroad business, however, has not improved as greatly as had been anticipated and the formation of the Association of American Railroads has apparently complicated the problem of non-official bodies. As a result, uncertainty existed until recently as to whether or not the associations would be able to hold the conventions in May. The officers of the minor mechanical associations looked to the Mechanical Division for advice, and this division, in turn, looked to the officers of the Association of American Railroads for guidance.

The officers of the minor mechanical associations were disturbed a few days ago to receive a letter from J. R. Downs, vice-president in charge of operation of the Association of American Railroads, in which he said:

"Our Association has not absorbed the associations enumerated in your letter, and from the standpoint of the activities of the Association of American Railroads, there is no necessity for holding any conventions such as you have enumerated.

"You are probably aware that, because of the neces-

sity for economy in railroad operation, the number of conventions being held by the various Divisions of our Association have been limited."

Apparently the Association of American Railroads has washed its hands of all responsibility for these organizations. Obviously the conventions are entirely off for the spring of this year. It would seem that some group somewhere, preferably the Mechanical Division, should step in and take the leadership in deciding the best course to follow. Should these associations be revived and be encouraged to resume their former activities and programs? How have changed conditions affected their importance and value? Will it be advantageous to combine some of them? Have some of them outgrown their usefulness—at least to a degree where their programs should be more or less drastically revamped? What assistance will be required to start committees to work to prepare for future conventions?

In such study as the *Railway Mechanical Engineer* made of this question last summer, and which was reflected in editorial comment in several issues, there is no question but what there is a real and substantial value in the work of most of these organizations. How can this value best be conserved?

Steam Locomotive Capacity and Loading

The steam locomotive is a highly flexible power unit. Designed for given conditions of train load and speed, it will continue to function with a surprising degree of effectiveness under loads far beyond those anticipated by the designer.

This characteristic of the steam locomotive, which has become well established during a period of expanding traffic and increasing train loads, has obscured another characteristic which will be of growing importance, particularly in passenger service, as train loads are gradually decreased by the introduction of coaches of lighter weight or by traffic conditions requiring fewer passenger cars per train. This is that maximum fuel economy is impossible with an under-loaded locomotive.

There are at least two reasons for this fact: First, the minimum combustion required to maintain a fire and keep up steam is in the nature of a fixed charge against the additional combustion required to move the load. The rate of this fixed charge increases as the load decreases. Second, the demand for power is a function of the total load moved. This includes the weight of the locomotive. Here the movement of the locomotive becomes a fixed charge against the movement of the train, the rate at which it is distributed increasing as the weight of the train decreases. The second consideration is probably in most cases of considerably greater importance than the first.

On a horsepower-hour basis the under-loaded locomotive may show satisfactory economy. Its increased proportion of weight so increases the horsepower hours delivered that when figured on the basis of the transportation work performed—that is, the car miles or the gross ton miles exclusive of locomotive tender—the performance is beyond the economical range.

With conditions developing which promise a decreasing trend in train weight it is important that new motive power, particularly for passenger service, be proportioned with this consideration in mind.

Steam Locomotive Accepts Challenge

Early in the present century enthusiastic advocates of the electrification of steam railroads predicted that the days of the steam locomotives were numbered. Indeed, according to some of their predictions, the steam locomotives would long before this have entirely disappeared, at least from main line service. Nevertheless, the steam locomotive is still with us, although electrification has been gradually extended, as conditions have warranted, and it is now possible to travel from Washington, D. C., to New Haven, Conn., by electrified service.

More recently another competitor to the steam locomotive has appeared in the form of the Diesel-electric type. Some of its advocates are even more enthusiastic than the electrical engineers of earlier days. Designers of the steam locomotive have accepted this new challenge. On January 2, the Chicago & North Western inaugurated its "400," a new train operating between Chicago and the Twin Cities and drawn by a converted 4-6-2 steam locomotive, using oil. This five-car train of standard equipment makes a run of 408½ miles in 420 minutes.

In the course of a few weeks the Baltimore & Ohio will place in operation two light-weight, non-articulated passenger trains. They will be propelled by two steam locomotives built for high-speed service—one four-coupled and the other six-coupled—in competition with a specially designed Diesel-electric locomotive. This should make possible the gathering of accurate data which will be helpful in determining the relative advantages and disadvantages of the different types.

The American Locomotive Company is now building a high-speed streamline locomotive for the Chicago, Milwaukee, St. Paul & Pacific, which has been carefully designed from the ground up for light, high-speed passenger service. The results will be watched with keen interest on the part of railroaders far and wide.

The steam locomotive is not dead yet by a long way and its proponents are determined to fight to the last ditch in demonstrating its merits, in comparison with other types of motive power.

What Is the Availability Of a Locomotive?

The availability of a locomotive for service has taken on particular significance since the Diesel engine entered the railroad picture. One of the claims of the proponents of Diesel power is that, due to the lack of necessity of much of the servicing required on steam power, it is ready for service a greater proportion of the time. In an editorial appearing in these columns last month figures were used, for comparative purposes, indicating an average availability of 88 per cent for the Diesel locomotive and 51 per cent for the steam switching locomotive. A correspondent, in discussing this question, takes exception to the figure of 51 per cent for the steam locomotive on the grounds that it is at least 10 per cent too low. He says that many times when the steam locomotive is ready for service but not "called," the entire time is charged against the locomotive and that, on the other hand, the Diesel switcher is usually credited with being available when actually it is not in use.

There is some question as to whether or not the average road keeps accurate records of the time that its steam locomotives are actually in service; out of service for running and general repairs, and available for service but not used. In view of the fact that new types of motive power are coming into the railroad field, both Diesel and steam, and that for purposes of comparing costs accurate records must be compiled it may be worth while to consider the desirability of collecting data which will show accurate comparisons between the several types of power. As to the question of comparative availability between the Diesel and steam it will probably be generally conceded that if for no other reason than that it requires relatively little servicing the Diesel has a higher availability than steam. In order to make accurate comparisons of availability, an accurate division of out-of-service time must be made between the time undergoing repairs or servicing and the time awaiting call. This latter should not be charged against the availability of any type of motive power.

Train Speeds In Retrospect

On July 12, 1834, the American Railway Journal, progenitor of *Railway Mechanical Engineer,* carried a short description of a race between one of the "new-fangled" steam trains and a horse-drawn stage, in which race the train passed the stage at a speed of 18 m.p.h. An indignant partisan of the older form of transportation used the next issue of the Journal to comment as follows: "Sir,— You certainly have made a mistake in saying that the coach, on Saturday last, went at 18 m.p.h., when it overtook and passed the stage, whose four horses were 'put to their utmost speed, with a velocity comparative to that with which the stagecoach would have passed a wagon.'

"Wyatt, of the Watford and Aylesbury coach, the one to which you refer, is very angry at your asserting that his fine team of horses were overtaken and passed in the manner you speak of by the steamer, when the latter did not perform more than 18 m.p.h. Wyatt knows the steamer well; he was once beaten by it going up Windmill-hill; and he says—and I say—and all the others say, that when the steam carriage overtook him, and passed him on Saturday, it was undoubtedly at a pace of more than 24 m.p.h.

"We have many times done two miles in five minutes, and you shall see it done again whenever you are so disposed.

"You show that the speed of the whole ten miles, including several stoppages and the turnpike, etc., was above 15 m.p.h. Surely to overtake and pass a team of four fine horses at their 'utmost speed,' as you did, must have required more than 18."

While the first steam trains proceeded at a very moderate rate, modern train speeds of 100 m.p.h. were at least contemplated at a very early date, as shown by the following quotation from an English paper which was reproduced in the Journal of July 26, 1834: "A speed of 40 m.p.h., with a light load, has been obtained upon the Manchester railway; and Mr. G. Stephenson, the engineer, has stated his opinion that an engine be constructed to run 100 miles within an hour, although he acknowledges that 'at that rapidity of motion the resistance of the atmosphere would be very considerable.' Engines are now made with eight times the power of the Rocket, yet with little more weight resting on each rail, the load being equally divided upon six wheels, and the machinery placed in a more advantageous situation than formerly. The tubes of the boiler are made smaller, and more numerous, and of brass instead of copper. The last engine put on the railway ran 23,000 miles with the most trivial repairs, making every day four or five journeys of 30 miles each."

NEW BOOKS

DOWMETAL. *The Dow Chemical Company, Midland, Mich. 68 pages, 8½ in. by 11 in. Paper bound. Price, $1.*

This data book explains the uses of Dowmetal, a name given to a group of magnesium-base alloys with a high magnesium content, and how this light structural metal may be fabricated by processes common in industry. A description is given of accepted shop practice; various available forms of Dowmetal are listed and described, such as sand or die castings, extruded shapes, forgings, sheet plate and strip; and methods of welding, riveting, forming, and machining are discussed in detail, with much information on designing and finishing.

With the Car Foremen and Inspectors

Convenient Device for Handling Truck Work

*By H. A. McConville**

Many different devices have been designed and built, the object of which was to facilitate handling the work on freight-car trucks, such as changing wheels and other parts. Among such devices, some of which have been described in the *Railway Mechanical Engineer* in the past, the one employed on the Louisville & Nashville at Montgomery, Ala., possesses certain features which make it superior to most others of a similar character.

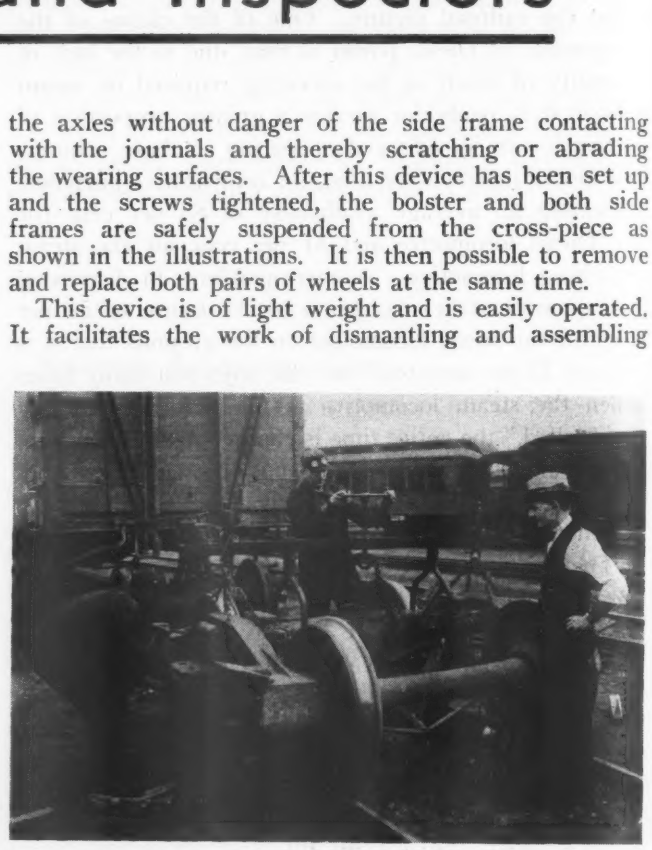
As will be noted from the illustrations this device is made up first of two upright supports or standards, each being formed from two pieces of 1¾-in. pipe, bent and welded together at the top to a rectangular pin. A horizontal cross-piece, carried by the supports, is made up of two ⅜-in. by 3-in. iron bars, welded at each end to ¾-in. by 3-in. bars which serve as tracks on which the side-frame lifters operate. At the center the cross-piece is strutted by a nut or tapped block through which the

Lifting a side frame for changing journal bearing

bolster lifting screw operates. The cross-piece is also provided with four pockets for receiving the rectangular pins on the upper ends of the supports. Trucks with ordinary bolsters are shown in the illustration, in which case the supports are placed inside the side frames and straddle the bolster. For trucks having extremely wide bolsters, or those having brake-beam safety attachments fastened to the spring planks and thus making it impossible to set up the standards in the inside position shown, the supports are then placed outside the truck, being fitted into the pockets located at the extreme ends of the cross-piece.

The raising and lowering of the bolsters and side frames is accomplished by means of screws. The side-frame lifters are mounted on wheels, the employment of which permits the side frames to be rolled off or onto the axles without danger of the side frame contacting with the journals and thereby scratching or abrading the wearing surfaces. After this device has been set up and the screws tightened, the bolster and both side frames are safely suspended from the cross-piece as shown in the illustrations. It is then possible to remove and replace both pairs of wheels at the same time.

This device is of light weight and is easily operated. It facilitates the work of dismantling and assembling



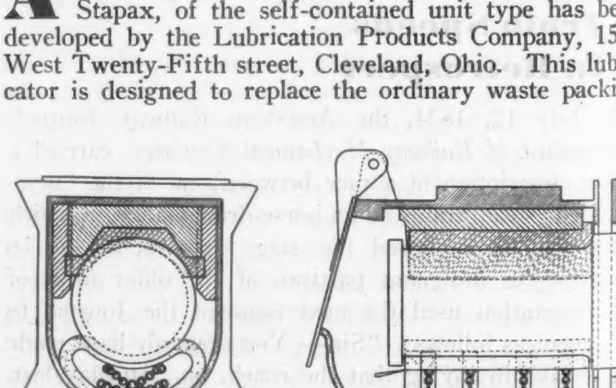
Lifting bolster for removal of springs

trucks, eliminates danger of personal injury and also serves as a positive protection to the axles in removing and replacing side frames, thereby minimizing the possibilities of cut journals after cars have been returned to service.

*Foreman car department, Louisville & Nashville, Montgomery, Ala.

Unit Type Journal-Box Lubricator

A NEW type of journal-box lubricator, known as Stapax, of the self-contained unit type has been developed by the Lubrication Products Company, 1531 West Twenty-Fifth street, Cleveland, Ohio. This lubricator is designed to replace the ordinary waste packing



Application of Stapax journal box lubricator

used in journal boxes of tenders, freight and passenger cars.

This lubricator is composed of a roll of heavy, high-grade wool felt and is controlled by light chain spiders that hold it in place and prevent its slipping and rolling up around the journal. Furthermore, there are no loose ends or free fibres in contact with the bearing. The object of the device is to supply positive lubrication at all times without waste of oil and to render unnecessary frequent inspection, attention and repacking. It is easily installed and is said to give long service besides minimizing the danger of hot boxes.

Referring to the illustration, it will be noted that two tape loops are provided on the outer end which are for use in removing the lubricator. For locomotive service the lubricator is provided with an extension lip on the inner end to assure lubrication of the hub liner which takes care of the lateral thrust. The Stapax lubricator is available also in special sizes suitable for any waste-packed box, such as those on overhead cranes.

Sandblasting a Passenger Car

THE illustration shows the operation of sandblasting an Illinois Central passenger coach No. 2294 preliminary to painting. This car is one of a number of cars which were thoroughly repaired, reconditioned and modernized as part of an extensive passenger-car rehabilitation program carried out by the Illinois Central during the past year.

In order to prepare the steel surfaces of the car for painting, the car was moved to the sandblast station shown in the illustration where four sandblast container tanks are located just outside the east end of the shop and equipped with four outlets for sandblasting. The hose extension shown in the illustration consists of a 1¼-in. sandblast rubber hose which may be equipped with various lengths of ¾-in. pipe nozzles, as required. These nozzles are made of extra-heavy pipe in order to avoid premature cutting out and frequent replacement of the nozzles.

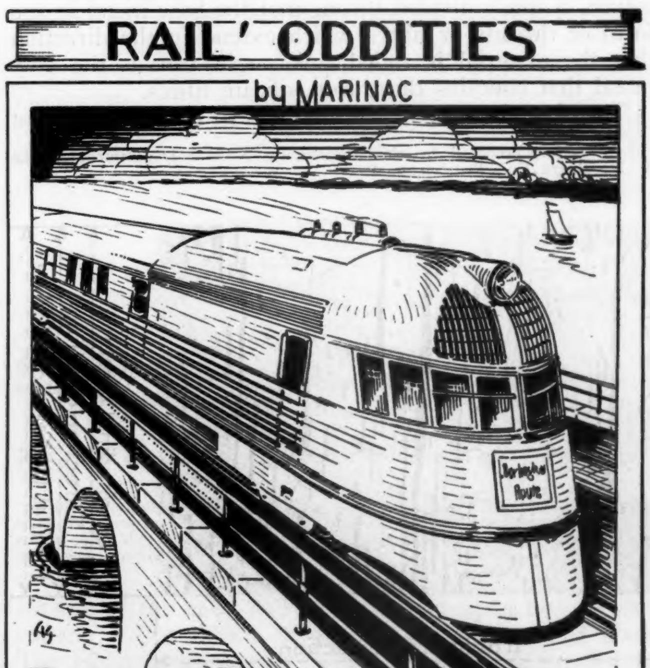
View of an Illinois Central passenger coach being sandblasted at Burnside shops

In preparing the car body for sandblasting, all parts are removed, including the sash. If, for any reason, certain sash must be retained in place, they are protected by means of metal shields. The shop pressure for exterior sandblasting is about 85 lb. and this pressure is also used for the interior of head-end cars. For the interiors of other cars, special sandblast equipment is used, which comprises ½-in. nozzles utilizing 35-lb. pressure.

A baggage car can be sandblasted, using the equipment illustrated, by two men in about eight hours, both exterior and interior, and including the roof and the steel underframe. On a coach exterior, it takes two men about 6½ hours. In accordance with customary practice, the sand is re-used as long as it retains its cutting properties.

New Use for Old Coaches.—The Great Western Railway of England has solved the problem which is bothering some railroads in this country—what to do with old passenger coaches. The Great Western is spotting them in a number of resort localities and renting them to camping parties. The coaches are completely equipped with cutlery, kitchen utensils, crockery, bed clothes, linen, oil lamps, mirrors and deck chairs, and they are for rent for periods of a week or more. Each coach has been planned to provide a two-berth and a four-berth sleeping compartment, as well as a large living room with a table, six chairs and a wardrobe, and a combined kitchen and scullery complete with sink, draining board, table, cupboard and a stove. Naturally the railway expects campers renting these cars to travel to and from the resorts by rail.

* * *



Further explanation furnished by the editor upon request

The Bending of Wrought Iron Plates*

SUCCESSFUL bending of any ductile plate material, such as wrought iron or steel, depends to a great extent upon the fabricator's knowledge of the structure of the metal, its physical characteristics, and the limitations imposed by the bending equipment available. Since many different methods are employed in bending plates, it is the purpose of this article to discuss in a clear and concise manner the ones that have been found most satisfactory for use when bending wrought-iron plates.

For almost a quarter of a century prior to 1930, plates made of wrought iron were not produced in commercial quantities. As a result, there are undoubtedly many fabricators today who are not thoroughly familiar with wrought iron and the differences between it and steel. For this reason, it is necessary to preface the discussion of plate bending principles and practices with a statement of two important facts and a brief description of wrought iron, its properties and characteristics.

1.—Wrought iron plates may be formed just as satisfactorily as steel plates if the fabricator observes the slight modifications in bending practice that are fully described in the following pages.

2.—The equipment employed in forming or bending wrought-iron plates is the same as that commonly used for steel plates. No special equipment is required.

Wrought Iron—Its Properties and Characteristics

Wrought iron is composed of two dissimilar materials, namely, (1) a high purity iron base metal, and (2) iron silicate or slag. The slag, a non-rusting, glass-like substance, is finely divided throughout the base metal in the form of threads or fibres which extend in the direction that the material is rolled. Wrought iron is the only metal that contains these iron silicate fibres.

The slag content, which amounts to about 3 per cent by weight of the total, confers on wrought iron a distinctly fibrous structure. This characteristic of the metal is very much in evidence when a specimen is nicked and fractured.

* An article prepared by the Engineering Service Department of the A. M. Byers Co., Pittsburgh, Pa.

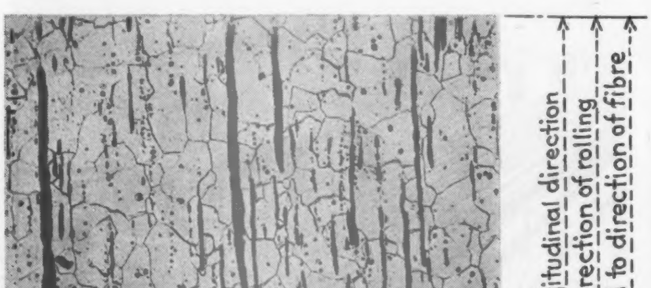


Fig. 1—Photomicrograph taken at 100 magnifications, showing structure of wrought-iron together with some of the terms used in the discussion of plate bending

The fibrous structure of wrought iron may, for the purpose of comparison, be likened to the structure of hickory wood, with which everyone is familiar. It is generally known that a hickory plank can be bent to a short radius without danger of breaking if the line of bend (or axis of bend) is at right angles to the grain. However, if the same plank is bent with the bend line parallel to the grain the radius of bend must be increased somewhat in order to keep the plank from splitting. The grain of the wood is responsible for this difference in "bendability."

The slag fibres in wrought iron affect its bendability in much the same manner that the grain affects the bendability of hickory wood. Therefore, it follows that a wrought-iron plate can be bent to a shorter radius when the line of bend (or axis of bend) is at right angles to the direction of the slag fibres than when the bend line is parallel to the fibre direction.

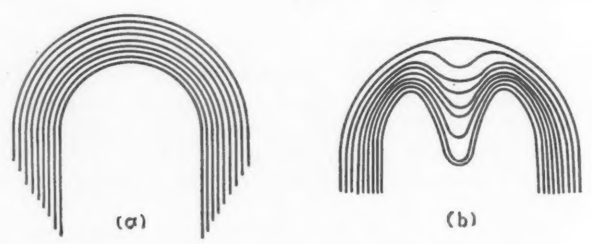


Fig. 2—(a) This illustrates what would happen if all sections of a plate were free to move—(b) When the ends are gripped tightly the inside fibers must buckle when bent unless the outside fibers stretch

From this comparison it is apparent that the slag fibres in wrought iron have a direct effect on the ductility of the material. This effect, however, is more noticeable in the transverse direction (across the plate) than it is in the longitudinal direction (lengthwise of the plate).

The reason for this difference in ductility in the two directions is that the slag fibres make the metal discontinuous in the transverse direction. This characteristic of the material is clearly illustrated by the photomicrograph shown in Fig. 1. This photomicrograph was taken at 100 magnifications and the slag fibres are most apparent. For the sake of clarity some of the terms used in this discussion of plate bending are indicated on Fig. 1.

Since ductility is the most important property of a material from the standpoint of "bendability" it should be mentioned that for standard wrought-iron plates the minimum ductility in the transverse direction is 2 per cent in 8 inches, while in the longitudinal direction the minimum ductility is 14 per cent in 8 inches. These values are somewhat less than those shown by other ductile non-slag-bearing metals, such as steel.

From the foregoing facts, it is evident that wrought iron is an entirely different material from any of the

other commonly used ferrous metals. Consequently, when bending wrought iron, its directional properties must be taken into consideration.

When wrought iron plates were reintroduced to the market in 1930, the A. M. Byers Company realized the limits that low transverse ductility might impose on the applications of the material. Consequently, extensive research and experiments led to the development of special rolling procedure which made possible the production of wrought-iron plates having much higher transverse ductility than those produced by rolling in the conventional manner. This development is particularly important because it has made possible the use of wrought-iron plates for many corrosive services where severe fabrication had formerly prevented their use.

Facts About Plate Bending

Most of the difficulties encountered in plate bending come through a lack of appreciation of what happens during the bending operation. When a plate is bent the metal must be stressed beyond its yield point. This means that the metal undergoes deformation or stretching. Therefore, since the ductility of the metal determines the amount it can be deformed before fracture occurs, it it obvious that ductility is the most important

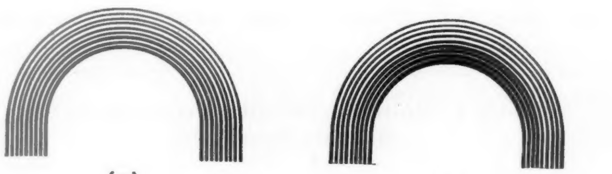


Fig. 3—(a) Illustrating the action of elastic parts each free to stretch individually—(b) If this same deck were gripped tightly at the ends and then bent the outer cards would thin out and the inner ones, instead of buckling, would compress and thicken.

factor to be considered. Recognition of this fact, and the use of a technique that takes it into account, is the first step in successful plate bending.

The part that ductility plays may be clearly shown by an example. Take a deck of cards (a magazine or a pad of paper will also serve) and bend it into a semicircle. It will take the form illustrated in Fig. 2(a).

From inside to outside, each successive card "steps back" a greater and greater distance from the edges of the inside card.

If the deck is gripped tightly at the ends to prevent slipping and then bent, it will look like the sketch in Fig. 2(b).

The cards on the inside, restrained from pushing down, must buckle to make room. The outer card will remain unchanged.

If the cards were elastic and could stretch, the "step-back" could be squared after bending by taking each individual card and drawing it down to the level of the inner one. The stretch required would increase with each card, reaching a maximum in the card on the outside of the curve. In order to permit the stretching, the cards would thin out in the center, and the deck would present the appearance shown by the sketch in Fig. 3(a).

If this elastic deck were tightly gripped at the ends, to prevent slipping, and then bent, the cards on the outer portion of the deck would thin out. The cards on the inner portion instead of buckling, would compress and thicken. The deck would then take the form illustrated in Fig. 3(b).

Somewhere near the center of the deck there would be one card that would be neither stretched nor compressed. This card would be located at what is commonly termed the neutral axis. The exact location of the neutral axis depends upon the type of plate material being bent. With the commonly used ductile ferrous plate metals, such as wrought-iron or steel, the neutral axis will be near the inner surface of the bend because those metals have greater strength in compression than in tension.

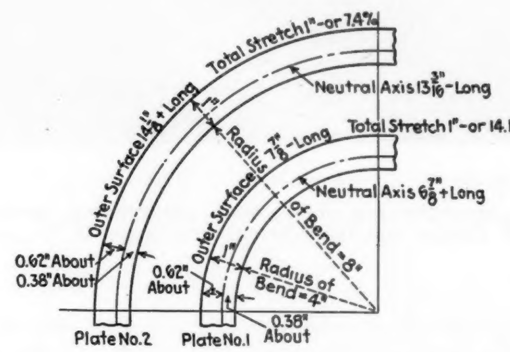


Fig. 4—The influence of the radius of bend on "bendability"

Again referring to Fig. 3(b), if all cards in the bent deck except the top and bottom ones were taken out, then those two cards might be considered as representing the inner and outer surfaces of a bent plate. Since the metal in a plate is a unit mass, it is obvious that slip cannot occur as it can between the cards when the deck is bent. When a plate is bent the necessary change in length of the two surfaces much be produced by a lengthening or stretching of the outer surface and (to a limited extent) to a shortening of the inner surface. However, for the purpose of this discussion the shortening of the inner surface that occurs during the bending operation can be neglected.

The importance of ductility and its influence on bendability has been mentioned. It follows then that if a plate is bent so as not to exceed the ductility of the metal, the amount of stretch that occurs on the outer

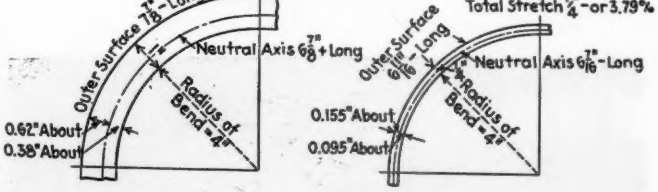


Fig. 5—The influence of plate thickness on "bendability"

Note:—A comparison of the two sketches reveals that the ¼-in. plate could be bent to a radius of 1 in. before the stretch in the outer surface would equal that which occurred in the outer surface of the 1-in. plate bent to a 4-in. radius.

surface of the bend will be dependent upon two factors, namely: (1) radius of bend; (2) thickness of plate. The effect of these two factors is clearly illustrated by the sketches in Fig. 4 and Fig. 5.

Fig. 4 shows two plates of the same thickness bent to different radii. A brief study of these sketches will reveal that the outer surface of each plate has stretched or elongated the same amount—slightly less than 1 in. The reason for this is that in each case the outer

Courtesy of the Walsh-Holyoke Steam Boiler Works, Holyoke, Mass.

Fig. 6—Wrought-iron plates ¾-in. thick being formed in a roll bender for fabrication into large O. D. pipe

surface of the plate is the same distance from the neutral axis. However, the neutral axis (which has the same length as the plate before it was bent) of plate No. 1 is appreciably shorter than the neutral axis of plate No. 2. Therefore, since the outer surfaces of the two plates have stretched the same amount, it is obvious that the percentage stretch per unit of length is much greater in Plate No. 1 than in Plate No. 2. This is clearly shown by the dimensions on the sketch.

Therefore, it can be stated that for a given plate thickness, the percentage stretch per unit of length increases as the radius of bend decreases.

The influence of plate thickness on bendability is clearly illustrated by Fig. 5 which shows two plates—one ¼-in. thick and the other 1-in. thick—bent to the same radius. It is apparent that during the bending operation the outer surface of the 1-in. plate had to stretch to a much greater extent than did the outer surface of the ¼-in. plate. Also, it will be observed that that ¼-in. plate could be bent to a much shorter radius than the 1-in. plate before the percentage stretch per unit of length in the outer surface would equal that which occurred in the outer surface of the 1-in. plate.

From these facts it can be stated that for a given radius of bend, the percentage of stretch per unit of length increases as the thickness of the plate increases.

Theoretically the degree of bend, that is, whether the bend is made through 45-deg., 90-deg., 180-deg., or 360-deg., has nothing to do with the amount that the material must stretch. From the practical standpoint, however, degree of bend does have an influence because many types of bending equipment used do not evenly distribute the stretch over the entire bent portion. For this reason the procedure followed and the equipment employed are factors that have a direct bearing on the bending of wrought-iron plates.

Courtesy of the Walsh-Holyoke Steam Boiler Works, Holyoke, Mass.

Fig. 7—Crimping the edge of a ¾-in. wrought-iron plate

Principles and Practice for Cold Bending of Wrought-Iron Plates

These recent developments in rolling procedure have made possible the production of wrought-iron plates possessing varying degrees of ductility in both the transverse and longitudinal directions. For example, wrought iron plates are available having a ductility in the transverse direction ranging from a minimum of 2 per cent elongation in 8 in. to a maximum of 8 per cent elongation in 8 in.

In the latest revision of the American Society for Testing Materials Specification for Wrought-Iron Plates, Designation A 42-34 T, provision is made for wrought-iron plates having varying degrees of ductility in the two directions. Table I, taken from this specification, shows the minimum ductility requirements for the various classes of plates available.

Table 1—Minimum Ductility Requirement for Wrought-Iron Plates

Transverse ductility (elongation in 8 in.) per cent		Minimum longitudinal properties: Tensile strength, min., lb. per sq. in.		Minimum longitudinal properties: Elongation in 8 in., min., per cent
2	51,000 — (1500 x 2) =	48 000*	(16 — 2) =	14*
3		46 500		13
4		45 000		12
5		43 500		11
6		42 000		10
7		40 500		9
8		39 000		8

*Test requirements for the usual full longitudinal rolling of the plate.

From this table it can be seen that as the transverse ductility is increased until the minimum requirement is 8 per cent in 8 in., the ductility as well as tensile strength in the longitudinal direction decreases. Using these values as a basis, a formula has been derived for calculating the recommended radii to which wrought-iron plates of various thicknesses can be bent in the two directions. This formula, which is given as follows, provides an ample factor of safety.

$$R = \frac{62\,T}{S} - .38\,T$$

where R = Minimum radius of bend in inches
T = Thickness of plate in inches
S = Per cent elongation in eight inches

Using this formula and substituting the value for ductility that will come into play when the bend is made, the minimum radius of bend has been calculated for the several classes of plates in the various thicknesses. These values are given in Table II.

These recommended bending radii were calculated using in each case the minimum specification requirements for ductility as shown in Table I. It will be observed that the sum of the minimum transverse ductility and the minimum longitudinal ductility is always sixteen per cent. Therefore, if a plate has a minimum

transverse ductility of 5 per cent, the minimum longitudinal ductility will automatically be 11 per cent.

The correct use of Table II necessitates a thorough knowledge of the relation between the bend line (axis of bend) and the direction of the fibre in the plate. For example, if a 1/4-in. standard rolled plate (minimum transverse ductility 2 per cent in 8 in., minimum longitudinal ductility 14 per cent in 8 in.) is to be bent with the bend line parallel to the direction of rolling, the recommended radius of bend will be 7 5/8 in. If the bend is made with the bend line at right angles to the direction of rolling, the minimum radius of bend will be 1 in. These values are obtained directly from Table II by referring to the columns opposite 1/4-in. plate headed 2 per cent and 14 per cent.

However, there are conditions where a standard rolled plate with 2 per cent transverse ductility and 14 per cent longitudinal ductility can not be used because the requirements demand a shorter radius bend than the 7 5/8 in. given in the preceding example. For such conditions it is obvious that the designer must select the type or class of plate that will bend to the desired radius. In other words, the plate must be specified in accordance with the ductility required to provide the necessary stretch that will occur when the bend is made.

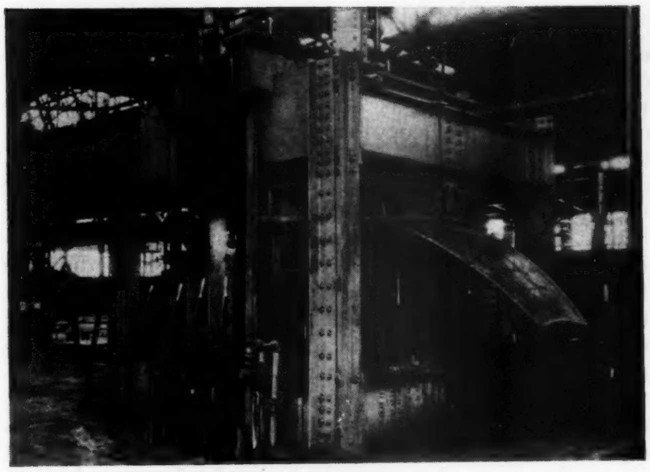
For example, if a 3/8-in. plate must be bent, with the bend line parallel to the length of the plate, to a 2 3/4-in. radius, it is readily seen that this can be done only with a plate having maximum transverse ductility (that is, a transverse ductility of 8 per cent), as shown in Table II under the column headed 8 per cent, opposite 3/8-in. plate.

It should be borne in mind that there are always to be found slight variations in the physical characteristics of a material, and that differences in bending practice and equipment have a pronounced influence on the results obtained. The recommended radii of bends given in Table II, as already mentioned, are sufficiently liberal to take ordinary variations in material, practice and equipment into account.

Methods and Equipment Used in Cold Bending

Since the bending of wrought iron plates necessitates the stretching of the material, it is obvious that distribution of the stretch or elongation is of the utmost importance. Good equipment or practice will result in distributing the stretch over the entire circumference of the bent portion. Poor equipment or practice may, in some rare cases, produce satisfactory results, but ordinarily the metal will be punished unnecessarily. Even within the physical limitations of the metal, poor equipment or practice can cause failure and consequent dissatisfaction.

For these reasons it is essential that we mention the various types of equipment and methods commonly employed in bending plates.



Courtesy of the Pittsburgh-Des Moines Steel Company, Pittsburgh, Pa.

Fig. 8—A ring type disher used for cold dishing heads up to 100 in. in diameter

Roll-Bending—Roll-bending is one of the most satisfactory methods to use when bending wrought-iron plates. The equipment employed permits the metal to stretch uniformly over the entire circumference of the bent portion. This, of course, is an important consideration when bending any plate metal. There are two types of roll-bending equipment in general use; namely, the pyramid roll and the initial roll. Fig. 6 shows a pyramid roll in operation bending 3/4-in. wrought-iron plates.

Courtesy of the Pittsburgh-Des Moines Steel Company, Pittsburgh, Pa.

Fig. 9—A spinning machine for cold flanging heads 28 in. in diameter and larger

Table II—Recommended Bending Radii for Wrought-Iron Plates

Plate thickness T	Values of S in per cent.												
	2	3	4	5	6	7	8	9	10	11	12	13	14
	Minimum Bending Radii in Inches—R												
3/16	5 3/4	3 11/16	2 13/16	2 1/4	1 7/8	1 9/16	1 3/8	1 1/4	1 1/16	1	1	13/16	3/4
1/4	7 5/8	5 1/16	3 3/4	3	2 1/2	2 3/16	1 11/16	1 5/8	1 7/16	1 5/16	1 3/16	1 1/8	1
5/16	9 9/16	6 5/16	4 3/4	3 3/4	3 1/8	2 5/8	2 5/16	2 1/16	1 13/16	1 5/8	1 1/2	1 3/8	1 1/4
3/8	11 1/2	7 5/8	5 11/16	4 1/2	3 3/4	3 3/16	2 3/4	2 7/16	2 3/16	2	1 13/16	1 5/8	1 1/2
7/16	13 1/4	8 7/8	6 5/8	5 1/4	4 3/8	3 11/16	3 1/4	2 7/8	2 9/16	2 5/16	2 1/8	1 15/16	1 3/4
1/2	15 5/16	10 1/8	7 9/16	6	5	4 1/4	3 11/16	3 1/4	2 15/16	2 5/8	2 3/8	2 3/16	2
5/8	19 1/8	12 11/16	9 7/16	7 1/2	6 1/4	5 5/16	4 5/8	4 1/16	3 5/8	3 5/16	3	2 3/4	2 5/16
3/4	22 11/16	15 3/16	11 5/16	9	7 7/16	6 3/8	5 1/2	4 7/8	4 3/8	3 15/16	3 9/16	3 5/16	3 1/16
7/8	26 11/16	17 3/4	13 1/4	10 1/2	8 11/16	7 7/16	6 7/16	5 11/16	5 1/16	4 5/8	4 3/16	3 13/16	3 9/16
1	30 5/8	20 5/16	15 1/8	12	9 15/16	8 1/2	7 3/8	6 1/2	5 13/16	5 1/4	4 13/16	4 3/8	4 1/16

Note 1—The radii given in this table were calculated to the nearest 1/16 in.
Note 2—These recommended bending radii are not to be taken as representing the absolute limit to which wrought iron plates can be bent. The values shown are sufficiently liberal to take into account ordinary variations in material, practice, and equipment.

V-Block and Hammer—The V-block and hammer is commonly used in plate bending. In general, this type of equipment is not so satisfactory for use in bending wrought-iron plates as some of the other types, because it has a tendency to localize or limit the stretch to a short segment of the bent portion. Obviously, if the amount of stretch exceeds the ductility of the material, fracture will occur on the outside of the bend. It should also be mentioned that in some cases the edges of the block or die are sharp enough to dig into the plate and tear the metal during the bending operation. In order to avoid this possibility, the edges of the die should be rounded sufficiently to permit the plate to move freely. Wrought-iron plates can be bent satisfactorily using a V-block and hammer, but the operator must be thoroughly familiar with the limitations of this type of equipment and with the characteristics of wrought iron. If a long plate is to be bent, the best results will be obtained if the complete bend is made in several passes rather than in one pass. In other words, the plate should be given an initial bend along its entire length in order to eliminate the possibility of causing a reverse kink or "buckle" in the unbent portion. The bend can then be completed in one or two passes.

Courtesy of the Pittsburgh-Des Moines Steel Company, Pittsburgh, Pa.

Fig. 10—Sectional flanging machines used for light gage heads

Gag-Press and Bulldozer—Wrought iron plates can be bent satisfactorily using either a gag-press or a bulldozer. Ordinarily, however, the bend should be made in several passes, thereby successively decreasing the radius so that the plate is not given a reverse bend. This is a particularly important consideration where a long plate is to be bent to a short radius.

Hot Bending of Wrought-Iron Plates

While a majority of the plate bending is done cold, there are certain types of bends where it is desirable to work the metal hot. Naturally, any metal can be bent more easily when hot, and wrought iron is no exception.

As in cold bending, the radius of bend is a controlling factor in hot-bending wrought-iron plates. In general the recommended bending radii shown in Table II can be decreased by approximately 50 per cent when hot bending is employed.

In hot bending the temperature at which the metal is worked is a most important factor. Wrought-iron plates should never be worked at a temperature in excess of 1,400 deg. F. Best results will be obtained if the temperature is around 1,350 deg. F. when the plate is bent.

At 1,350 deg.-1,400 deg. F. wrought-iron plates will have a dull red color in the average lighted shop. It is always advisable to eye-gage the temperature after the plate has been placed in the die by looking underneath to see the color. This procedure will give a more nearly accurate idea as to the exact temperature.

If the temperature is to be measured with an optical pyrometer, every precaution should be taken to knock the scale off the surface of the metal before the reading is taken. If this is not done, an erroneous reading will ordinarily result, because the scale will be several hundred degrees cooler than the metal.

Compound Bending of Wrought Iron Plates

Compound bending, as the term would indicate, means that the plate metal is bent in more than one direction.

It is beyond the scope of this article to do more than to touch on this phase of plate bending because there are so many variables involved.

Wrought iron plates can be easily formed either hot or cold by practically any of the commonly used methods provided, of course, that the characteristics of the material are taken into account. One of the most important considerations is to employ a method that will permit the stretch in the metal to be as nearly uniform as possible. Localized stretching should always be avoided.

Flanged and Dished Heads for Tanks

Special wrought-iron plate stock is produced for forming flanged and dished heads. With this special head stock it is possible to produce heads in accordance with the A. S. M. E. requirements, which state that the knuckle radius shall be not less than three times the plate thickness, or 6 per cent of the diameter of the head. Wherever possible, the largest allowable radius should be used.

Flanged and dished heads are formed by either spinning or pressing, depending upon the size. The metal is worked either hot or cold. Large heads made of heavy-gage metal are formed hot on a spinning machine and with wrought-iron plates, the best results will be obtained if the metal is worked at a temperature of 1,350 deg.-1,400 deg. F.

Peripheral speed is an important consideration in hot-flanging large wrought-iron heads on a spinning machine. It has been found that a speed of about 800 ft. per minute gives good results. This is a somewhat slower speed than that ordinarily used for steel, but, generally speaking, wrought iron must be worked more slowly than steel if any severe forming operation is to be entirely satisfactory.

Cold dishing and flanging of wrought-iron heads can be easily accomplished if ordinary precautions are taken. Figs. 8, 9 and 10 show types of equipment that can be used for cold dishing and flanging wrought-iron heads.

De-scaling Wrought-Iron Plates

For some applications it is necessary to remove the mill scale from plate metal in order to provide a smooth surface for painting or other purposes. There are several different ways of removing scale, but acid pickling and blasting are the most commonly used.

The scale formed on wrought-iron plates during the rolling operations adheres tightly to the surface of the metal. In the majority of services for which wrought-iron plates are used, it would be desirable to leave the scale in place.

Pickling—When removing scale from wrought-iron plates by means of acid pickling, the pickling bath should be maintained in an active condition with an acid concentration of at least 5 to 6 per cent. The bath should be maintained at a temperature of not less than 120 deg. F. Under these conditions the scale will be completely removed in a short time without undue injury to the underlying metal. In some cases it may be desirable to use an inhibitor in the acid bath to produce a smoother finished surface. However, if the scale is removed in order that the surface may be given a metallic coating, such as galvanizing, the use of an inhibitor is not advisable. When wrought iron is pickled, the acid attacks the metal, but not the slag, thereby making the resultant surface comparatively rough. Obviously a rough surface offers a better anchorage for a coating than a smooth one, and for that reason wrought iron will take on a heavier and more adherent coating than will other metals.

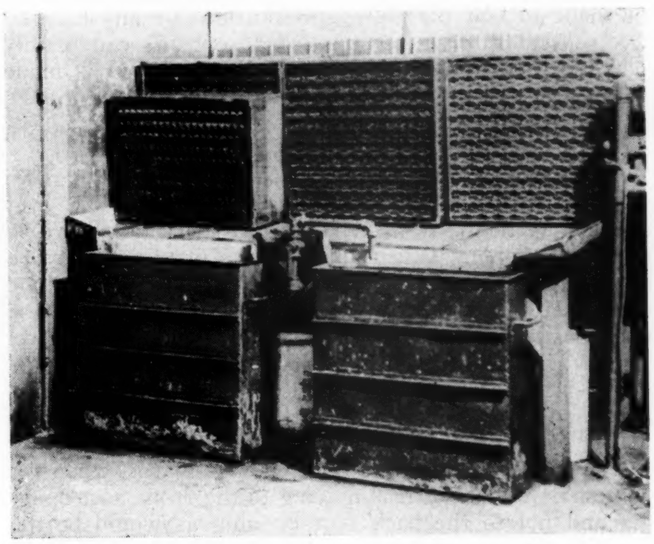
Sand and Metal Blasting—Sand blasting has been used for years to remove scale from metal surfaces. Within the past few years, however, another method of blasting has been developed in which hardened metal particles are employed in place of sand. Generally speaking, blasting with metal particles has a more severe effect on the plate material than does sand. This is particularly true of wrought iron because it is soft and has a fibrous structure. Where it is necessary to remove the scale from wrought-iron plate by blasting, we recommend that sand blasting be used. This method can be relied upon to remove completely the scale, and it does not seriously affect the metal surface.

Cleaning Filters-Testing for Freon Leaks

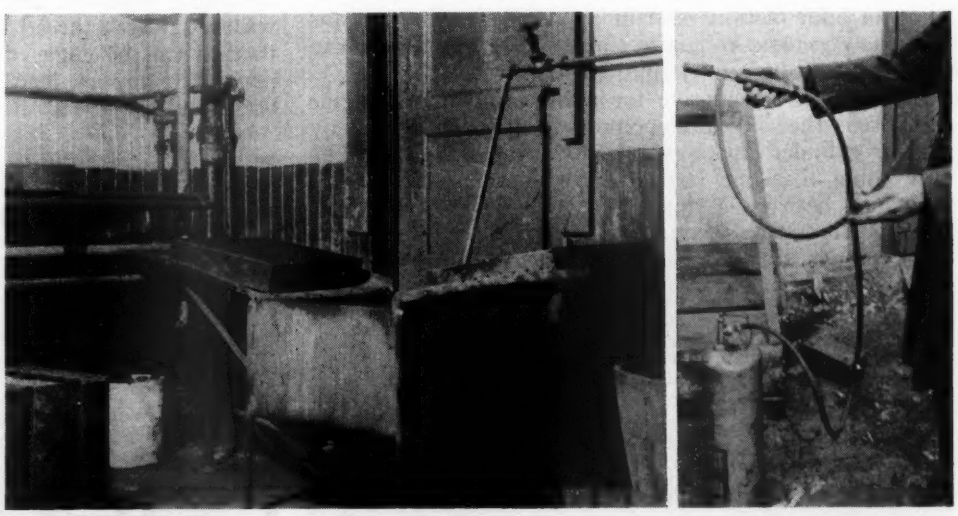
TWO maintenance operations now assuming considerable importance in connection with the operation of air-conditioning equipment are the satisfactory cleaning of filters and the testing for Freon leaks in those mechanical systems which utilize this type of refrigerant.

In cleaning the filters, a jet of compressed air is first blown through the filters in a direction reverse to the normal air flow to remove loose dust and dirt. The filters are then immersed in a tank of tri-sodium-phosphate which is kept boiling by means of a steam jet. They are then removed from this tank and placed on the sloping drain tables located on either side of the tank, as shown in one of the illustrations. The filters are placed on steam-heated horizontal iron pipes, after which they are immersed in clean oil in the vertical cans shown in the second illustration. Drain boards are provided behind these cans and serve to permit the excess oil to drain back into the cans.

The third view, given at the right in the first illustration, shows an interesting device for testing air-conditioning equipment for Freon leaks. A Prest-O-Lite tank is provided with a torch burner of the Bunsen type. A small tube attached to the burner draws in air to mix with the Prest-O-Lite gas. If the end of this tube, shown in the operator's left hand, is placed in the vicinity of a leak, any Freon gas drawn in with the air causes a characteristic change in the color of the Bunsen flame. This forms a convenient testing device.



Clean oil immersion tanks and filter drain pan



Filter cleaning tank and dry racks — (Right) Device for detecting Freon gas leaks

In the Back Shop and Enginehouse

Valve-Setting Indicator And Blow Chart*

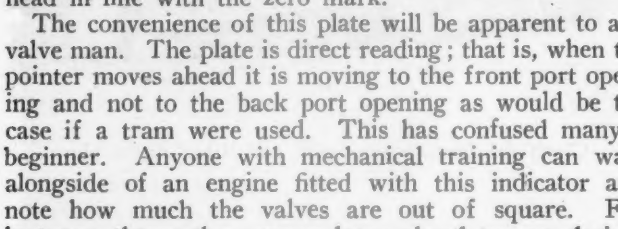
A VALVE-setting indicator and blow chart similar to that shown in the illustration can easily be made to suit the valve specifications of any locomotive. With it valve settings for all cut-offs can readily be observed. The particular chart, as shown, is made up for the following valve specifications: Valve travel, $8\frac{5}{8}$ in.; lap, $1\frac{5}{8}$ in.; lead, $\frac{1}{4}$ in.; exhaust clearance, $\frac{1}{16}$ in. inside admission; Baker valve motion.

To assemble this indicator on an engine use the valve tram in the usual way and place the valve on dead center. With the tram make a clear mark on the valve crosshead guide. When applying the plate to the valve crosshead guide be sure that this mark is directly line and line with the zero mark on the plate. After the plate has been fastened, drill two holes and insert dowel pins. The indicating pointer should be made of $\frac{1}{2}$-in. by $\frac{1}{4}$-in. steel and electric welded to the valve crosshead in line with the zero mark.

The convenience of this plate will be apparent to any valve man. The plate is direct reading; that is, when the pointer moves ahead it is moving to the front port opening and not to the back port opening as would be the case if a tram were used. This has confused many a beginner. Anyone with mechanical training can walk alongside of an engine fitted with this indicator and note how much the valves are out of square. For instance, throw the reverse lever ahead to any desired cut-off and move the engine forward. We will say the pointer has moved ahead to $3\frac{3}{16}$ in. and back to $3\frac{5}{16}$ in., and in the back motion the pointer moved ahead to $3\frac{5}{16}$ in. and back to $3\frac{3}{16}$ in. It will be noted that there is an error in both motions of $\frac{1}{8}$ in. This error being in both motions and in opposite directions means that the eccentric rod is at fault. The alterations can then be made by any desired method.

To test the locomotive for a blow move the engine and place the indicator pointer so it will locate any ring or cylinder packing blow noted on the plate. The only thing that one has to know is the difference in the sound between a ring and a cylinder packing blow. During thirty-five years of experience the writer has seen engines stripped of all the heads simply to find a blow and this same thing is still being done today.

* This device, which was designed by an old valve setter, is in use on a number of engines of a large railroad.



Valve position indicator and blow detector

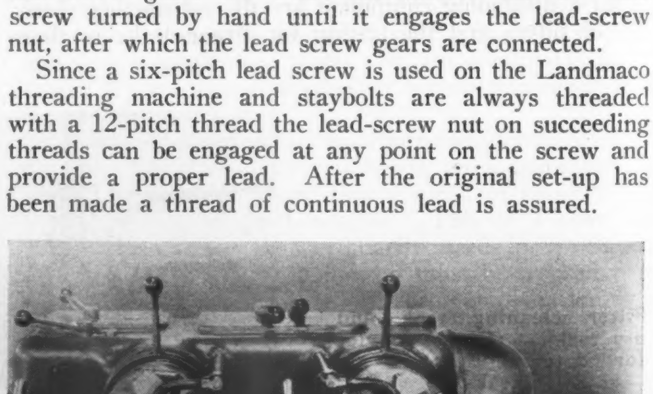
Timing Attachment for Staybolt Threaders

THE Landis Machine Company, Inc., Waynesboro, Pa., has recently placed on the market an attachment for use on their Landmaco threading machines for threading reduced body staybolts with a continuous pitch thread on both ends.

The timing attachment consists of a bracket attached to the carriage end cover and provided with positions for holding the gage bracket as may be required for various lengths of staybolts. The bracket arm attached to the gage bracket carries the gage for timing the thread. The arm being pivoted in the bracket permits it to be moved readily for clearing or engaging the thread. Refering to the illustration, on the right-hand side of the machine the gage is swung back to permit removal or insertion of the staybolt in the vise, while on the left the gage is about to be engaged in the thread.

The usual practice is to thread one end of the staybolt in the usual manner on one side of the machine without the use of the timing attachment. The set-up for the other end is made with a master staybolt which has been properly threaded with a continuous pitch thread on both ends. The master staybolt is inserted in the die head and the chasers closed to engage the thread. The timing attachment bracket is attached in its proper position for the length of the staybolt and the carriage set so that the timing attachment gage will engage the thread on the back end of the master staybolt. The lead-screw gears are then disconnected and the lead screw turned by hand until it engages the lead-screw nut, after which the lead screw gears are connected.

Since a six-pitch lead screw is used on the Landmaco threading machine and staybolts are always threaded with a 12-pitch thread the lead-screw nut on succeeding threads can be engaged at any point on the screw and provide a proper lead. After the original set-up has been made a thread of continuous lead is assured.



Timing attachment for Landmaco staybolt threader

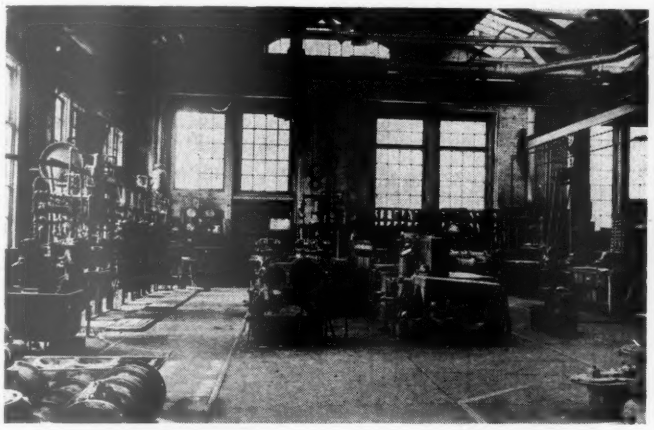
Air Compressor Repairs At Pitcairn Shop*

WITH few exceptions the 4,800 steam locomotives owned by the Pennsylvania are equipped with 9½-in. single-stage and 8½-in. cross-compound air compressors. When the failure of an air compressor to function requires its removal from a locomotive it is the established policy of the Pennsylvania to return that compressor to a central repair shop where it is completely overhauled and replaced in stock for reapplication. Central air-compressor repair departments are maintained at Pitcairn, Pa.; Wilmington, Del.; Altoona, Pa., and Ft. Wayne, Ind. The methods used for overhauling air compressors at the Pitcairn air-brake repair shop, which are described in this article, are typical of those used at the aforementioned central repair points on the system. The practices involved in this work are covered by a set of P. R. R. standard instructions which detail the manner in which the repair and testing of these units must be carried out by all repair shops. An average of thirty 9½-in. compressors and seventy-five 8½-in. cross-compound compressors are repaired at Pitcairn each month.

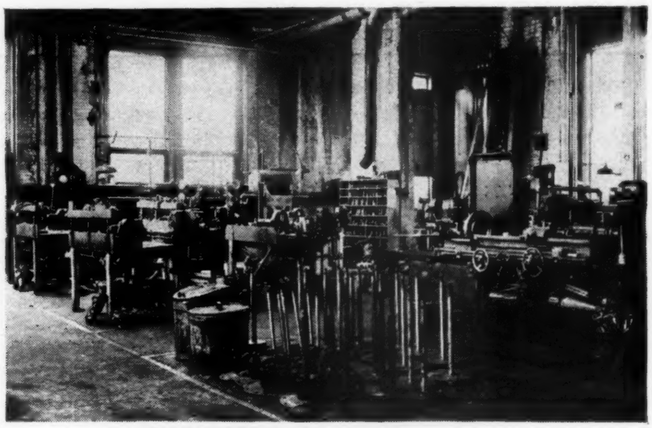
Before an air compressor is detached from a locomotive the air inlet strainers, drain cocks and lubricators, together with the piping and fittings, are removed and, when it is necessary to return any of these parts to the central shop for repairs, they are packed in such a manner as to preclude any damage in transit. All of the openings in the air compressor are plugged with standard pipe plugs upon removal from a locomotive and wooden blocks are applied to the compressor bolting lugs. The air-compressor repair work at the Pitcairn air-brake repair shop is concentrated in the air-pump repair section at the southwest corner of the shop. The facilities used in this work are shown in the shop layout drawing which appeared on page 116 of the March issue of the *Railway Mechanical Engineer,* as well as in the illustrations accompanying this article.

The compressors, when received at Pitcairn, are first given a thorough cleaning, before dismantling, in the lye vats located just outside the shop building in order to remove all grease and dirt. After being thoroughly cleaned the pump is suspended over the lye vat and operated by means of an air line which is coupled to the steam end to force the lye solution out of the pump and back into the vat. While being thus operated the air end of the pump is washed out with clean water and then drained. After this the pumps are placed on a small truck and taken into the shop where they are lifted by means of a two-ton overhead crane on to a special rack preparatory to being given a general inspection. This general inspection includes the application of whiting in order to discover cracked cylinders, broken lugs or cracks in any other of the pump castings. A test under 130 lb. air pressure is next made for leaking stuffing boxes and porosity so that any such defects may be found before the pump is overhauled and reassembled.

The pumps are then placed on another special rack designed to facilitate dismantling. Immediately upon dismantling all of the pump parts are cleaned with a non-inflammable turpentine substitute, special attention being given to the cylinders which are swabbed out with this cleaning solution. Metallic rod packing is used and the date the packing is applied, when done at the time of general overhauling, is stencilled on the center casting. This packing is carefully inspected and, if the marking on the center casting shows that it has been in service less than one year, and is in good condition it is left in the stuffing box for further service.



The air compressor repair section at Pitcairn



Parts repair benches in the air compressor repair section

Repair and Inspection Methods

The practices described above apply generally to both 9½-in. single-stage and 8½-in. cross-compound air com-

The lye vat with adequate handling facilities is located outside the shop near the repair section

* The second article dealing with the work of the Pennsylvania air-brake repair shop at Pitcairn, Pa. The first article appeared on page 115 of the March issue.

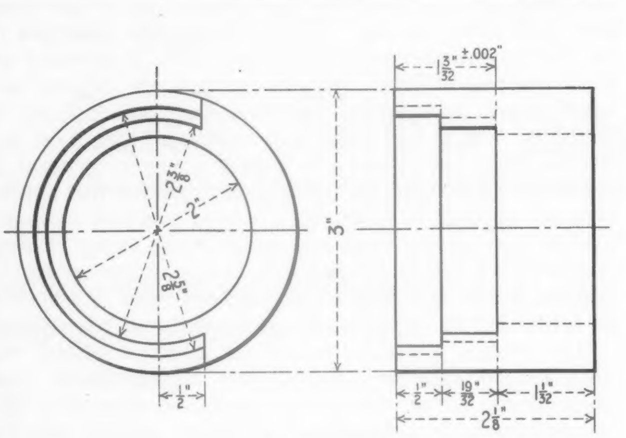


Fig. 1

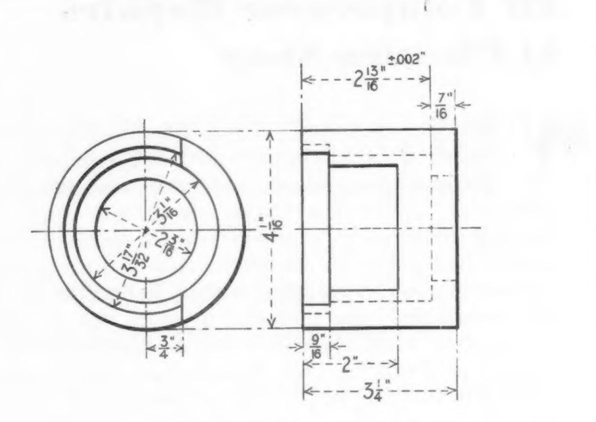
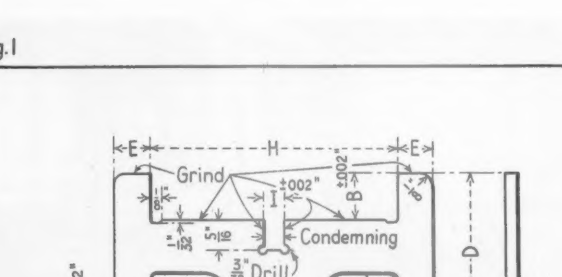


Fig. 2



No. Req'd	Dimensions in Inches								
	A	B	C	D	E	F	G	H	I
1	.212	$\frac{1}{2}$	$\frac{9}{16}$	$1\frac{5}{8}$	$\frac{3}{8}$	$\frac{3}{4}$	$3\frac{3}{8}$	$2\frac{5}{8}$	$\frac{7}{32}$
1	.400	$\frac{11}{16}$	$\frac{3}{4}$	$2\frac{1}{8}$	$\frac{1}{2}$	$1\frac{3}{16}$	$4\frac{1}{2}$	$3\frac{1}{2}$	$\frac{1}{4}$

Fig. 3

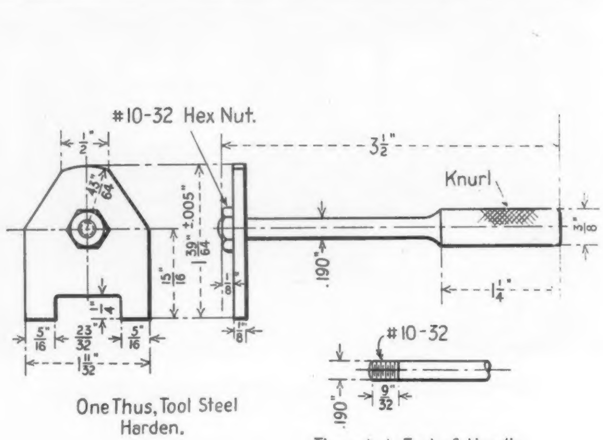
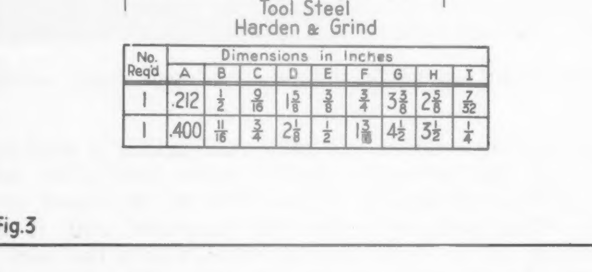


Fig. 4



No. Req'd	A	Stamp Thus.
1	.317"	When Gage Enters, Regroove to Next Size
1	.323"	" " " " " " "
1	.329"	" " " " " " "
1	.335"	" " " " " " "
1	.341"	" " " " " " "
1	.347"	" " " " " " "
1	.351"	" " " , Scrap Piston.

Fig. 5

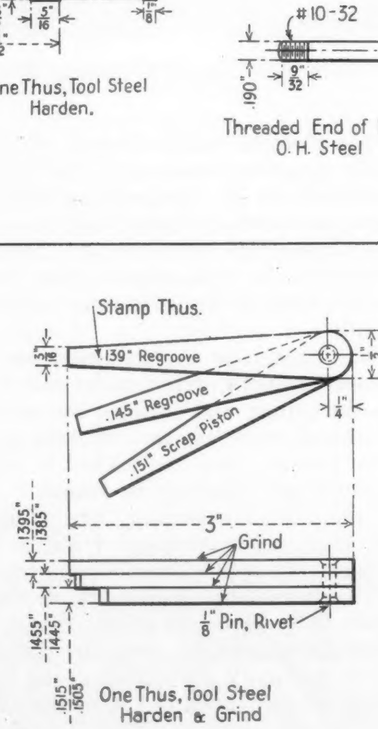


Fig. 6

Fig. 1—Lift gage for upper intermediate valve

Fig. 2—Fixture for determining lift of upper inlet valve

Fig. 3—Condemning and checking gage for upper intermediate, discharge and inlet valve cap

Fig. 4—Condemning gage for reversing valve bushing

Fig. 5—Condemning gage for steam and air piston ring grooves

Fig. 6—Condemning gage for main and exhaust valve ring grooves

pressors. While there is a difference in the parts of these two types of pumps, this article will go into detail on the repair practices and gages used in overhauling an 8½-in. cross-compound compressor. Certain of the gages mentioned are used on both 9½-in. and 8½-in. compressor parts, and the data on steam and air cylinders relate to both types of compressors. The practices described in connection with 8½-in. compressors are typical of all air-compressor repair work.

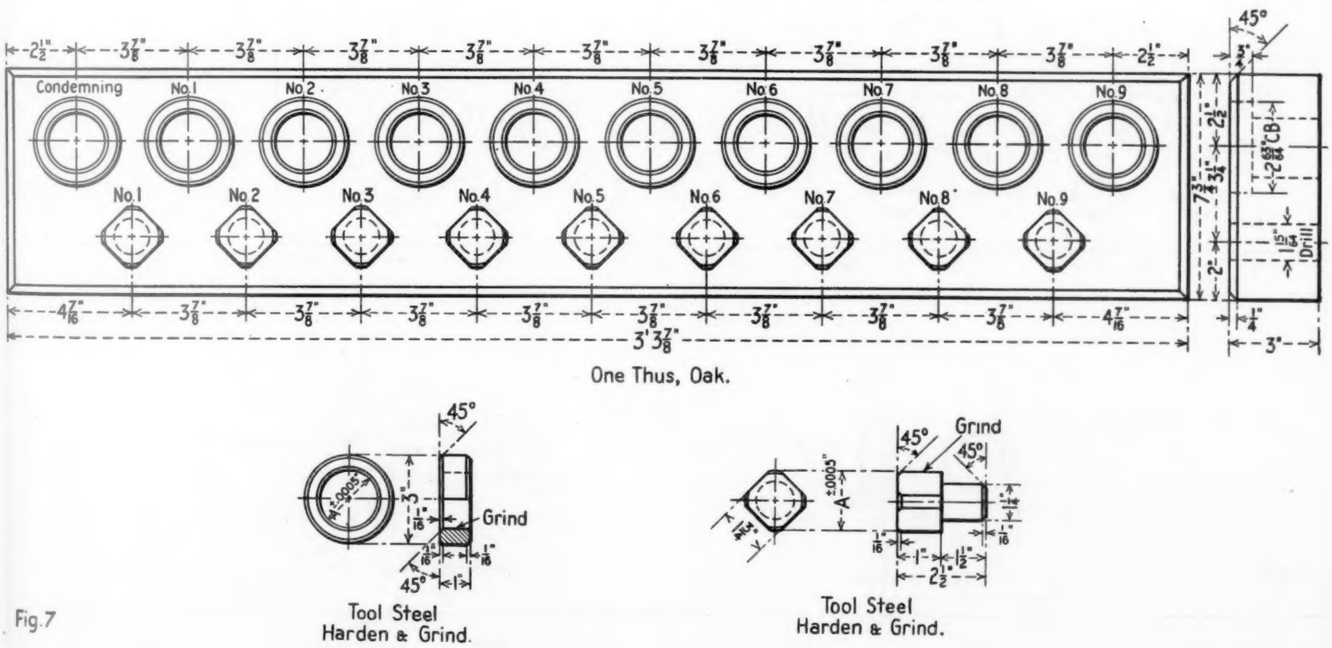
Air and Steam Cylinder Heads and Center Castings.—After the compressors are dismantled and the parts have been thoroughly cleaned, one of the first jobs to be done is to "mike" the steam and air cylinders to determine whether or not they must be rebored. Unless it is absolutely necessary, the steam and air cylinders are never separated from the center casting. The limits for air and steam-cylinder wear on both 9½-in. and 8½-in. cross-compound compressors are shown in Table I, and if the cylinders are worn $\frac{1}{16}$ in. or more beyond the standard diameters shown in the table, or when there is a difference of $\frac{1}{32}$ in. between the largest and smallest diameter of a cylinder, the cylinder must be rebored. When cylinders are not worn to the limits which require reboring, piston heads are applied that are not more than $\frac{1}{16}$ in. smaller in diameter than the largest diameter of the cylinder. New piston heads and rings of increased diameter are applied when cylinders have been rebored to the sizes shown in Table I. When renewing the high-pressure air pistons of the 8½-in. cross-compound compressor, steel pistons are used. On the final reboring of cylinders to 9¾ in. (in the case of 9½-in. pumps) the counterbore at the ends of the cylinder should not exceed $9\frac{13}{16}$ in., especially for the steam cylinder, as this would leave the margin too small to make a tight joint between the steam ports and the bore of the cylinder. The face of the cylinder is turned to provide a uniform surface for the top head gasket in order to insure a steam-tight joint at this point for the top head.

When the steam or air cylinders of an 8½-in. cross-compound compressor are worn $\frac{1}{16}$ in. larger in diameter than the maximum or final rebore, as shown in column 3 of Table I, they are rebushed and bored to the standard diameter shown in column 1. When this is done the bridge in the ports of the high-pressure air cylinder are cut out before the bushing is applied. The by-pass grooves in the low-pressure steam cylinder are maintained in accordance with Fig. 21.

One of the principal jobs that must be done on a center casting is to inspect it for poor threads and leaking stuffing boxes. When threads are discovered that are worn beyond serviceability, the threaded openings are built up by welding, drilled and retapped to the original standard. The inspection for leaking stuffing boxes is made, as previously mentioned, at the time the air pump first enters the shop. When leaking stuffing boxes are found by the inspector, they are removed and the stuffing-box seat in the center casting is refaced with a special tool. New stuffing boxes are applied if necessary in order to remedy the trouble, and when this is done they are doweled in place. When the clearance between

Table I—Sizes for Reboring Air Compressor Cylinders

Kind of compressor	Cylinders to be rebored	1 Diameter when new, in.	2 Diameter after first rebore, in.	3 Diameter after second or final rebore, in.
9½ in.	Steam cylinder	9½	9⅝	9¾
9½ in.	Air cylinder	9½	9⅝	9¾
8½ in. c.c.	High pressure steam cylinder	8½	8⅝	8¾
8½ in. c.c.	Low pressure air cylinder....	14½	14⅝	14¾
8½ in. c.c.	Low pressure steam cylinder.	14½	14⅝	14¾
8½ in. c.c.	High pressure air cylinder...	9	9⅛	9¼



Gage No.	No. Req'd.	A Diameter	Stamp Thus
No. 1	1	2.000 in.	2.000 in. — No. 1
No. 2	1	2.005 in.	2.005 in. — No. 2
No. 3	1	2.010 in.	2.010 in. — No. 3
No. 4	1	2.015 in.	2.015 in. — No. 4
No. 5	1	2.020 in.	2.020 in. — No. 5
No. 6	1	2.025 in.	2.025 in. — No. 6
No. 7	1	2.030 in.	2.030 in. — No. 7
No. 8	1	2.035 in.	2.035 in. — No. 8
No. 9	1	2.040 in.	2.040 in. — No. 9

Gage No.	No. Req'd.	A Diameter	Stamp Thus
	1	1.990 in.	1.990 in. — Condemning
No. 1	1	1.995 in.	1.995 in. — No. 1
No. 2	1	2.000 in.	2.000 in. — No. 2
No. 3	1	2.005 in.	2.005 in. — No. 3
No. 4	1	2.010 in.	2.010 in. — No. 4
No. 5	1	2.015 in.	2.015 in. — No. 5
No. 6	1	2.020 in.	2.020 in. — No. 6
No. 7	1	2.025 in.	2.025 in. — No. 7
No. 8	1	2.030 in.	2.030 in. — No. 8
No. 9	1	2.040 in.	2.035 in. — No. 9

Fig. 7—Gages for selecting the sizes of upper inlet and discharge valves and seats and lower inlet valve gages

the piston rod and the stuffing box exceeds .035 in., a new gland is applied.

Top Head Inspection and Repair.—The main valve piston in the top head is inspected for worn parts, those most frequently requiring replacement being the piston rings in the large and small piston sections and the exhaust section of the main piston valve. When the ring grooves in the large and small main-valve piston or in the exhaust piston are worn .002 in., they are trued up and rings .006 or .012 in. thicker than standard are applied. The gage shown in Fig. 6 is used for determining when the ring grooves in the main piston should be trued up. When fitting rings to large and small main valve pistons, also the exhaust piston, the minimum clearance is not less than .0015 in. or more than .003 in. When the large or small main-valve piston bushings or the exhaust bushings are worn or ground to a size $\frac{3}{64}$ in. larger in diameter than standard, they are renewed. When the top head is reassembled gaskets are applied under the large and small piston-valve cylinder covers and all top heads of $8\frac{1}{2}$-in. cross-compound compressors are equipped with eye-bolts for lifting. When the cap screw holes for the large and small piston-valve cylinder covers in the top head are worn the threaded holes are drilled and tapped and new bronze threaded bushings applied, restoring cap screw holes to the original size. Any cracks in the head casting are welded. This also applies to any other cracked castings in the air pump. When the reversing valve in an $8\frac{1}{2}$-in. cross-compound compressor is worn so that the exhaust cavity is less than $\frac{7}{32}$ in. deep, it is renewed, and when worn wedge-shape, it is trued up. A broaching tool is used for truing up the seat in the reversing valve bushing without the removal of the bushing from the head. The gage shown in Fig. 12 is used to determine whether or not the reversing valve is worn to the condemning limit. When fitting reversing valves to reversing valve bushings a clearance of .005 in. minimum to .032 in. maximum is maintained.

When the distance from the slide-valve seat to the top of the reversing-valve bushing exceeds $1\frac{39}{64}$ in., or when the thickness of the bushing where the reversing-valve chamber cap makes its seal is less than $\frac{3}{64}$ in., the bushing is renewed. The gage shown in Fig. 4 is used to determine whether or not the reversing-valve bushings are worn to the condemning limits.

When fitting reversing-valve cap nuts care is taken to assure that a bearing is obtained on both the head casting and the upper end of the reversing-valve bushing. A special tool has been developed for facing the bearing surface on the top head casting.

When the threads in the top head for the reversing-valve chamber cap fit are worn beyond serviceable limits the head casting is bored out and a bronze bushing containing threads of standard size is applied. The condemning and checking gage for reversing-valve chamber caps is shown in Fig. 10.

Reversing-valve rod lengths are checked with a standard gage shown in Fig. 20. A tolerance of .010 in. is allowed on new rods and .031 in. on old rods. A maximum tolerance between reversing-valve rods and bushings of .010 in. is permitted.

Air and Steam Piston Heads and Rods.—Air-compressor piston heads are condemned primarily for outer diameter and ring-groove wear. The gage shown in Fig. 5 is used to determine the condition of the ring grooves. When any of these grooves in air and steam pistons are worn .003 in. over any standard step size they are trued up, and rings .006 in., .012 in., .018 in.,

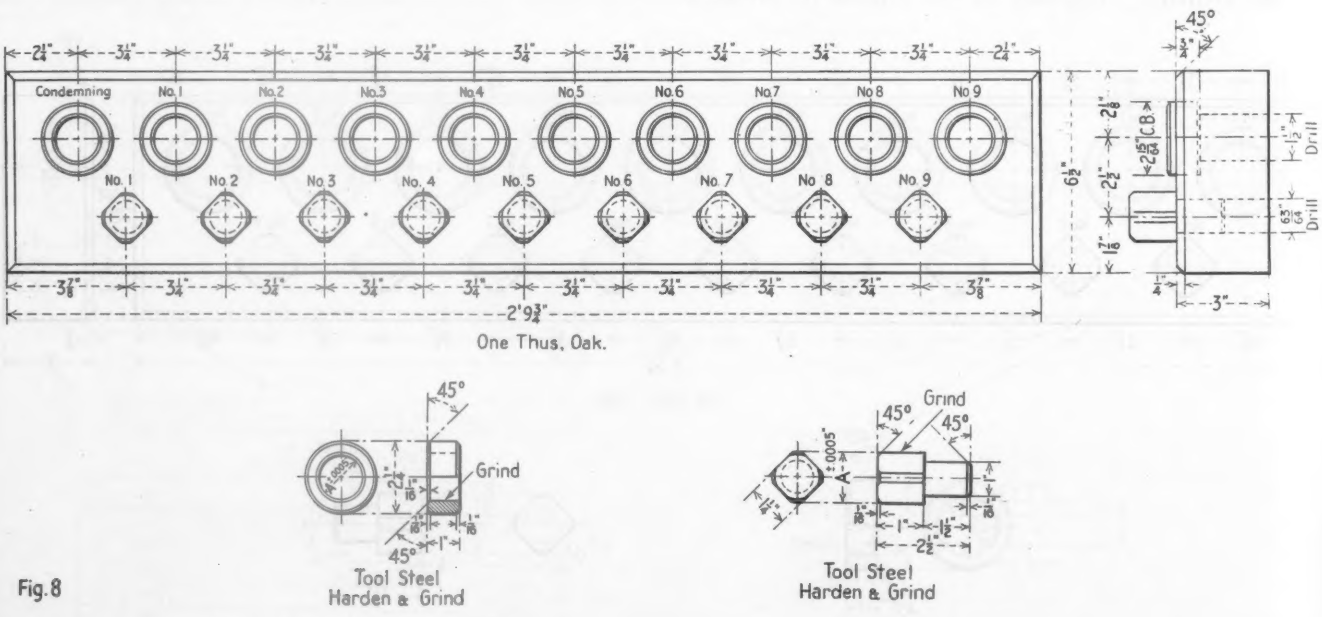


Fig. 8

Gage No.	A Diameter	Stamp Thus	No. Req'd.	Gage No.	A Diameter	Stamp Thus
No. 1	1.500 in.	No. 1 — 1.500 in.	1		1.490 in.	1.490 in. — Condemning
No. 2	1.505 in.	No. 2 — 1.505 in.	1	No. 1	1.495 in.	No. 1 — 1.495 in.
No. 3	1.510 in.	No. 3 — 1.510 in.	1	No. 2	1.500 in.	No. 2 — 1.500 in.
No. 4	1.515 in.	No. 4 — 1.515 in.	1	No. 3	1.505 in.	No. 3 — 1.505 in.
No. 5	1.520 in.	No. 5 — 1.520 in.	1	No. 4	1.510 in.	No. 4 — 1.510 in.
No. 6	1.525 in.	No. 6 — 1.525 in.	1	No. 5	1.515 in.	No. 5 — 1.515 in.
No. 7	1.530 in.	No. 7 — 1.530 in.	1	No. 6	1.520 in.	No. 6 — 1.520 in.
No. 8	1.535 in.	No. 8 — 1.535 in.	1	No. 7	1.525 in.	No. 7 — 1.525 in.
No. 9	1.540 in.	No. 9 — 1.540 in.	1	No. 8	1.530 in.	No. 8 — 1.530 in.
			1	No. 9	1.535 in.	No. 9 — 1.535 in.

Fig. 8—Gages for selecting the sizes of upper intermediate valves and valve seats and lower intermediate valves and cages

.024 in., .030 in. or .036 in. thicker than standard are applied. In fitting rings in air and steam pistons a clearance of not less than .002 in. nor more than .005 in. is maintained. When there is a difference of .002 in. in the diameter of the piston rod for metallic packing and .010 in. for soft packing the rod is trued up on a piston-rod grinder. When steam piston rods are worn or ground 5/64 in. below the original standard size, the rod and the pistons are scrapped. The piston and rod assemblies are checked to make sure that the standard distance between steam and air pistons is maintained, the distance being as shown in Table II.

Inspection and Replacement of Air Valves and Valve Parts.—In repairing air compressors a vitally important part of the job is the careful inspection and checking of air valves, seats, cages and caps. In order to maintain standards which have been set up the Pennsylvania has developed a rather elaborate set of gages for checking air valves and valve parts. Generally speaking, there are five kinds of these gages for use on an 8½-in. cross-compound compressor; namely, those for (a) selecting the proper sizes of valves, seats and cages; (b) determining the lift of valves; (c) checking valve seats and caps; (d) condemning valve caps and cages, and (e) determining the height of stop on valves. Reference to

Table II—Distances between Steam and Air Pistons

Size compressor	Standard	Minimum
8½ in. c.c.	22 11/16 in.	22 43/64 in.
9½ in.	18 11/16 in.	18 43/64 in.

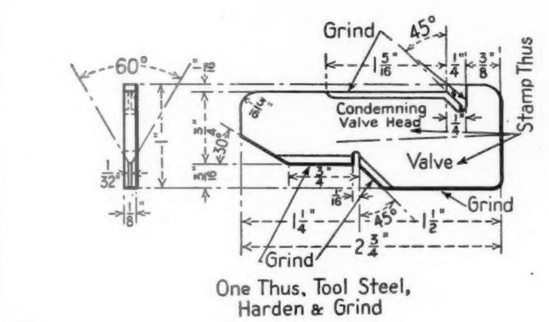


Fig.9

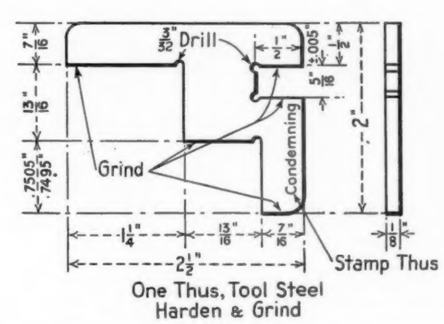


Fig.10

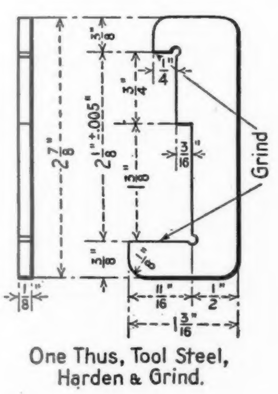


Fig.11.

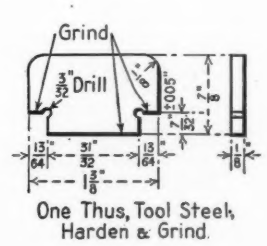
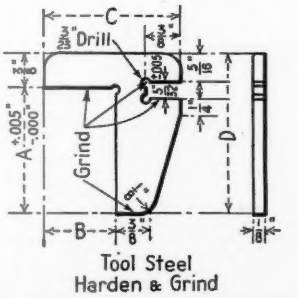
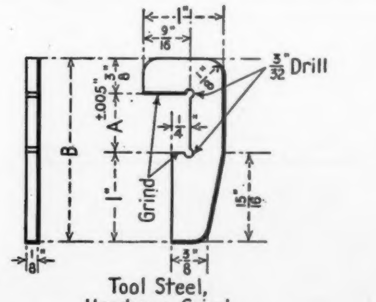


Fig.12



No. Req'd.	Dimension in Inches				Identification
	A	B	C	D	
1	1 5/8"	1/2"	1 5/16"	2	Condemning Cage, Lower Intermediate Valve Cage.
1	1 15/16"	1/2"	1 3/16"	2 5/16"	Condemning Cage Lower Inlet Valve Cage

Fig.13



No. Req'd	Inches		Identification
	A	B	
1	19/32"	1 31/32"	Checking Cage Upper Intermediate Valve Seat
1	23/32"	2 3/32"	Checking Cage Upper Discharge Valve Seat

Fig.14

Fig. 9—Condemning and checking gage for valves, seats and cages
Fig. 10—Condemning and checking gage for reversing valve chamber cap
Fig. 11—Checking gage for upper inlet valve seat
Fig. 12—Condemning gage for reversing valve
Fig. 13—Condemning gage for lower intermediate and inlet valve cages
Fig. 14—Checking gage for upper intermediate and discharge valve seats

the valves and parts and a key to the figure number of the gages used for the various operations is shown in Table III, the figure numbers referring to the drawings accompanying this article. Many of the valves and parts are common to both 9½-in. single-stage and 8½-in. cross-compound compressors, and several of the gages are, therefore, applicable to both types of pumps. References to the piece numbers shown in the table will simplify the determination of those parts which are common to both types. A standard valve lift of $\frac{3}{32}$ in. is maintained and, when fitting valves to cages or seats, the tolerance for the wings in the cage or seat is maintained at not less than .005 in. or more than .010 in.

The work in the air-pump section relating to compressor valves is all handled at one place in the section by men who specialize in that class of work. All air compressors on P. R. R. locomotives are equipped with steel air-valve cages and seats. The threaded portions of these parts are coated with a mixture of graphite and oil before being applied. When steel air-valve cages or seats are worn they are trued up and fitted with oversize valves, the references to gages used for the selection of these parts being shown in Table III. Only sufficient metal is removed from the cage or seat to true

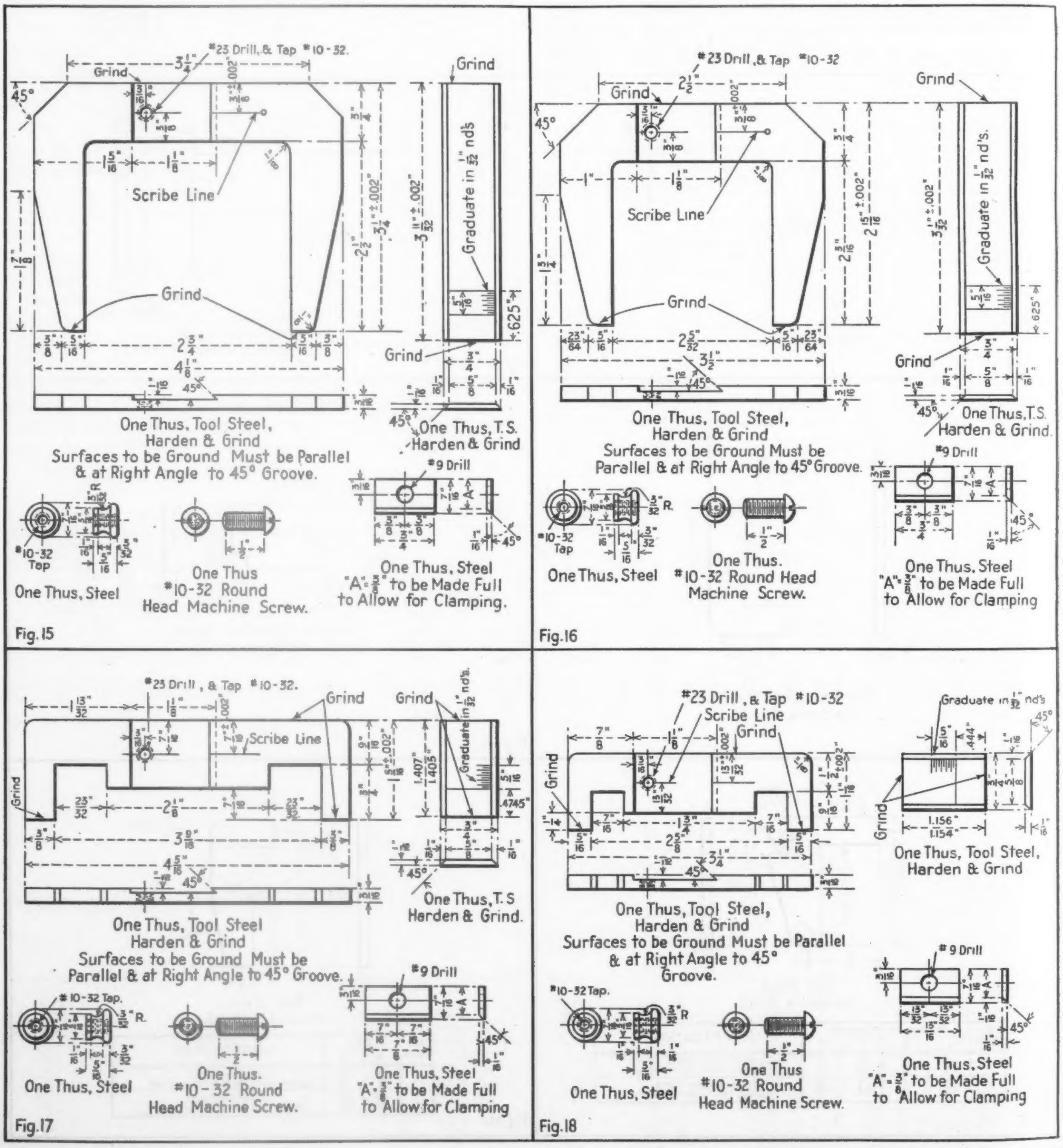


Fig. 15—Gage for determining lift of lower inlet valve cage

Fig. 16—Gage for determining lift of lower intermediate valve cage

Fig. 17—Gage for determining lift of upper inlet and discharge valves

Fig. 18—Gage for determining lift of upper intermediate valve

Table III—P. R. R. Standard Gages for Compressor Air Valves and Valve Parts

Part name and location	W.A.B. Co. Piece No.	1 For selecting sizes	2 For determining lift	3 Checking gages	4 Condemning gages	5 For determining height of stop
Valve, inlet, upper	29177	Fig. 7	Fig. 2 Fig. 17	Fig. 9	Fig. 9	Fig. 19
Valve, inlet, lower	29177	Fig. 7	Fig. 15	Fig. 9	Fig. 9	Fig. 19
Valve, intermediate, upper	24396*	Fig. 8	Fig. 1 Fig. 18	Fig. 9	Fig. 9	Fig. 19
Valve, intermediate, lower	24396†	Fig. 8	Fig. 16	Fig. 9	Fig. 9	Fig. 19
Valve, discharge, upper	29177	Fig. 7	Fig. 17	Fig. 9	Fig. 9	Fig. 19
Valve, discharge, lower	29177	Fig. 7	Fig. 15	Fig. 9	Fig. 9	Fig. 19
Valve seat, upper inlet	77117	Fig. 7	......	Fig. 11	......	No stop
Valve seat, upper discharge	8269	Fig. 7	......	Fig. 14	......	No stop
Valve seat, upper intermediate ..	8430*	Fig. 8	......	Fig. 14	......	No stop
Valve cap, upper inlet	1697	Fig. 3, Line 2	Fig. 3, Line 2	Fig. 3, Line 2	Fig. 3, Line 2	Fig. 3, Line 2
Valve cap, upper discharge	1697	Fig. 3, Line 2	Fig. 3, Line 2	Fig. 3, Line 2	Fig. 3, Line 2	Fig. 3, Line 2
Valve cap, upper intermediate ..	1906*	Fig. 3, Line 1	Fig. 3, Line 1	Fig. 3, Line 1	Fig. 3, Line 1	Fig. 3, Line 1
Valve cage, lower inlet	44445	Fig. 7	Fig. 15	Fig. 9	Fig. 13	Fig. 15
Valve cage, lower intermediate..	46806†	Fig. 8	Fig. 16	Fig. 9	Fig. 13	Fig. 16
Valve cage, lower discharge.....	44445	Fig. 7	Fig. 15	Fig. 9		Fig. 15

The gages shown in the table for the piece numbers marked * are also applicable to the upper inlet valves (piece No. 24,396); valve seats (piece No. 8,430), and valve caps (piece No. 1,906) of the 9½-in. single-stage compressor.

The gages shown in the table for the piece numbers marked † are also applicable to the lower inlet and discharge valves (piece No. 24,396) and valve cages (piece No. 46,806) of the 9½-in. single-stage compressor.

it up. Air valves are turned or ground to fit the cage or seat.

Final Testing and Inspection

After the compressor has been repaired the workman's number, the date repaired and the place repaired are stencilled on the upper front rim of the compressor center casting with ¼-in. letters, the stencil appearing, for example, as follows: PC–B.–L.–375–2/21/35, in which "PC" indicates Pitcairn shop; "B.," the initial of the mechanic who assembled the pump; "L.," the initial of the inspector on the final inspection; "375," the serial number of the repair job, and the last numerals the date the pump was turned out of the shop. If metallic packing is applied the date of application is stencilled on the center casting, as already indicated.

Table IV—Test Specifications for Air End of Compressors

Type of compressor	Boiler pressure not less than, lb.	Speed of compressor, single strokes per minute	Air pressure to be maintained in main reservoir, lb.	Diameter of orifice, in.
9½ in.	100	120	59	3/16
8½ in. c.c.	130	100	53	5/16

After the compressors have been assembled, which work is done at one station in the section by a specialist and a helper, and before they are sent out to be placed in service, they are tested on a specially equipped test rack. This rack, shown in one of the illustrations, has a capacity of five 8½-in. cross-compound compressors and three 9½-in. compressors. A main steam-line pressure of 140 to 150 lb. is available at the rack, and the test rack is so piped that the air from the pumps undergoing test on the rack may be fed back into the shop line. When the rack is filled to capacity with pumps undergoing test there is sufficient air capacity to supply the needs of the air-brake shop should this be necessary by reason of failure of the power plant.

Table V—Compressor Speed Used In Testing Steam Ends of Compressors

Pounds steam pressure	Single strokes per minute 9½ in.	8½ in. c.c.
100	140	40
110	150	64
120	160	90
130	170	110
140	178	124
150	182	134
160	190	148
170	194	156
180	198	164
190	202	168
200	206	172

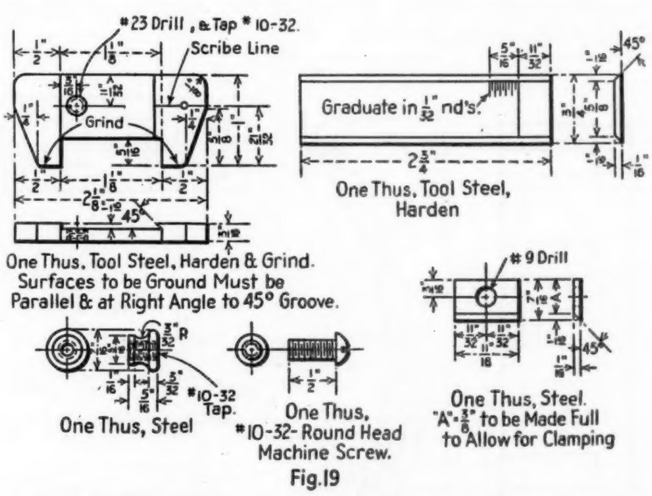
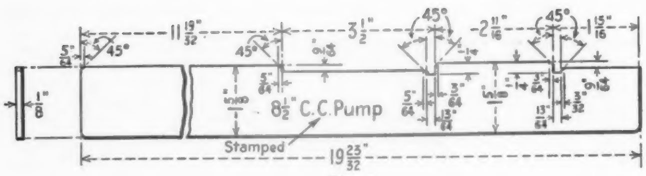


Fig. 19—Gage for determining the height of stop of air compressor valves

After a compressor is placed on the test rack the steam and air ends are well lubricated and the compressor is run a sufficient length of time to insure that the various new parts have found a bearing. Before making the final test the main reservoir and its connections, comprising a portion of the test rack, are tested by stopping the compressor after reaching a predetermined air pressure, say 100 lb., and noting the amount of leakage, which must be less than 2 lb. per min. In order to determine the condition of the air end of the compressor a disc with an orifice suitable for the type of compressor to be tested, as shown in Table IV, is located in a cock in the pipe leading from the main reservoir to the atmosphere. The compressor to be tested is then started and the steam supply throttled to make a specific number of strokes per minute. When the pressure in the main reservoir has been raised to within 10 lb. of that specified, the cock holding the disc is opened to the atmosphere. With the disc valve open the compressor must raise the pressure in the reservoir to that shown in Table IV with the given number of strokes per minute.

In testing the steam end of compressors the compressor steam throttle is opened wide and the main reservoir



Steel

Fig.20

Fig. 20—Gage for checking the length of reversing-valve rods

pressure is regulated by a cock in the main reservoir leading to the atmosphere from the main reservoir to 59 lb. for the 9½-in. compressor and 53 lb. for the 8½-in. cross-compound compressor. When these conditions have been established the steam pressure and compressor speed in single strokes per minute are observed and compared with those shown in Table V. Under these conditions the speed of the compressor for a given steam pressure must not be less than the number of single strokes shown in Table V.

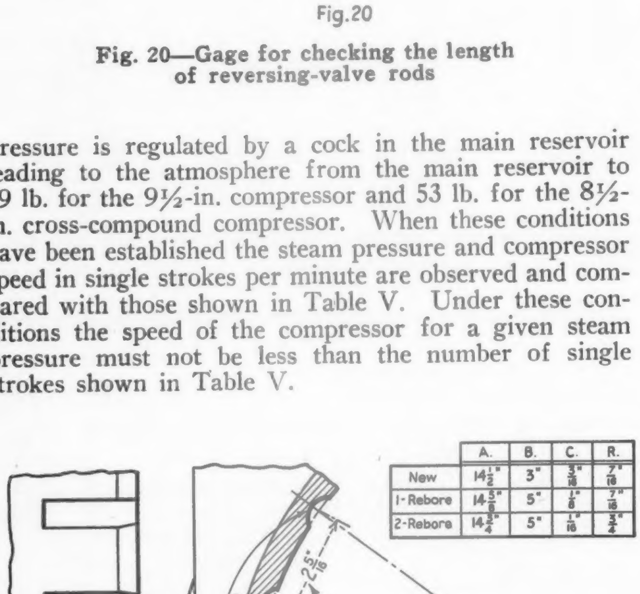


Fig. 21—Location of by-pass grooves in low-pressure steam cylinders

After a compressor is removed from the test rack a protector is applied to the lugs of both steam and air cylinders and all openings in the compressor are closed with standard pipe plugs. The compressor is equipped with a type B lubricator bracket at the repair shop, but the application of the Type B lubricator, drain cocks, strainers or fittings is not permitted until after the compressor has been applied to the locomotive.

Babbitt Metal*

THE mechanical and engineering departments of practically every worth-while railroad system impose rigid specifications upon bearing metal manufacturers in strict accordance with which all babbitt metal requirements are purchased.

The argument put forth in defense of these rigid specifications is that the performance and mileage cost of all journal boxes and lined bearings is directly dependent upon the performance of the babbitt liner because basically it controls frictional heat and thus serves to minimize hot box annoyance, particularly when lubrication for some reason or other is temporarily inadequate.

Now, in order to insure the ultimate in anti-friction qualities the babbitt specifications are so written as to eliminate certain elements which are calculated to promote frictional heat. In this connection is a lead-base lining metal, copper in excess of certain prescribed limits, and zinc in any amount, are specifically eliminated. The reason for the restriction on copper is that lead and copper are not affinities—lead repels the copper and will therefore unite beyond certain proportions only under special conditions of alloying. That accounts for the excess copper separating from the balance of the lead-base babbitt mixture and segregating in the form of hard spots.

Copper also tends to make the babbitt more thickly liquid and increase the grain size considerably in lead-base babbitts. Large crystalline grain indicates a loose or open structure which will more readily break down under a heavy load or severe service.

Zinc is literally "poison" in a lead-base babbitt alloy. It, too, will unite with lead and stay in solid solution only in small proportions. But as little as one-eighth of one per cent of zinc will utterly ruin an otherwise good lead-base babbitt alloy. Zinc, like copper, has little affinity for lead and when zinc is introduced into lead-base babbitt metal it quickly causes the molten babbitt to become a more or less "mushy" or drossy mass out of which it is extremely difficult to pour anything that remotely resembles the good bearing liners which the engineering department had in mind when they wrote the specifications for the lining metal.

Zinc is a hard "dry" metal thoroughly unsuited for bearing purposes particularly when tangled up with a lot of lead-base babbitt in which it segregates and, combining as it does with some copper, tin and antimony, forms hard spots in the bearing liner. Zinc being of a lighter specific gravity than lead, will rise and float on the surface when segregation occurs. It should, therefore, not be difficult to understand that since the surface metal in a pouring ladle, for instance, is the first to pass over the lip and into the journal box casting, the hard zinc mixture as a consequence is sure to find its way into the bearing liners.

But in the face of these facts I have observed in several railroad shops on my visits, babbitt metal pots filled with molten lining metal so contaminated with zinc and copper that the entire surface of the molten metal was covered over with a blue, purple and reddish-colored metallic "scum," perhaps one-half inch in thickness. It should be borne in mind that this metallic "scum" on the surface of the molten metal is simply the excess zinc and copper, together with some antimony, tin and lead, beyond the saturation point of lead for zinc and copper, particularly.

Practically all brass castings are alloys composed of copper, lead, tin and zinc. The average run of scrap car brasses removed from foreign equipment when it comes into the shop for repairs, contains approximately three to four per cent of zinc, four to six per cent of tin, and 70 to 75 per cent of copper, the balance being lead with about one per cent of impurities.

When babbitt chips become contaminated with brass borings or chips, do not invite a headache—condemn the material and sell it so that it will be sure not to get into the babbitt pots.

*From a paper presented before the November meeting of the Southern & Southwestern Railway Club by A. Fritschle, sales manager, Federated Metals Corp., St. Louis, Mo.

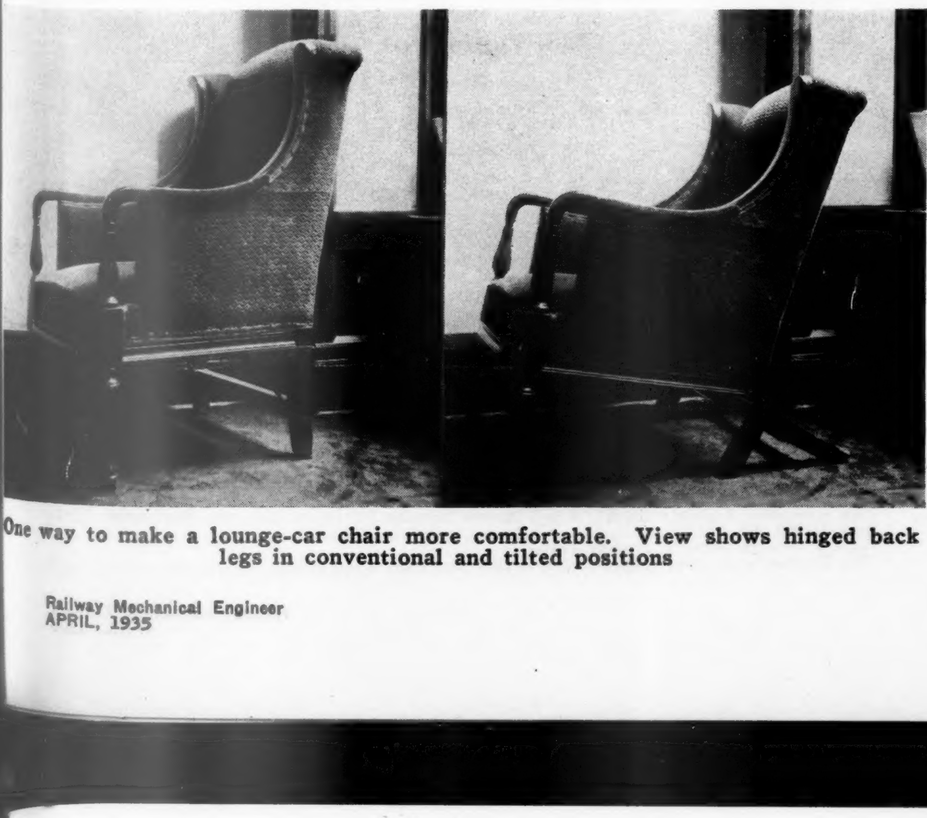
Among the

Clubs and Associations

WESTERN RAILWAY CLUB.—W. G. Black, vice-president of the Chesapeake & Ohio, will discuss "Good Transportation Depends on Efficient Motive Power" at the meeting of the Western Railway Club to be held on April 15 at 8 p. m. at the Hotel Sherman, Chicago.

CANADIAN RAILWAY CLUB.—"Has the Economic Performance and Maintenance of Locomotives and Cars Failed To Meet the Adverse and Changing Conditions of the Post-War Period?" was the topic presented by J. R. Macken, supervisor of costs, office of chief of motive power and rolling stock, Canadian Pacific, before the April 8 meeting of the Club.

MECHANICAL DEPARTMENT CONVENTIONS.—Last fall tentative arrangements were made looking toward the holding of three-day conventions of eight of the mechanical department associations during the early part of May. While these associations are not a part of the Association of American Railroads, they have been looking for guidance and advice to the officers of that association. ¶ Many of the officers and members of these associations, including the Air Brake Association, American Railway Tool Foremen's Association, Car Department Officers Association, International Railway Fuel Association, International Railway General Foremen's Association, International Railway Master Blacksmiths' Association, Master Boiler Makers' Association and Traveling Engineer's Association, were sorely disappointed a short time ago to receive a letter from J. R. Downes, vice-president of the Association of American Railroads, in which he made this statement: "Our Association has not absorbed the associations enumerated in your letter, and from the standpoint of the activities of the Association of American Railroads, there is no necessity for holding any conventions such as you have enumerated. ¶ "You are probably aware that, because of the necessity for economy in railroad operation, the number of conventions being held by the various divisions of our Association have been limited."



One way to make a lounge-car chair more comfortable. View shows hinged back legs in conventional and tilted positions

Directory

The following list gives names of Secretaries, dates of next regular meetings and places of meeting of mechanical associations and railroad clubs:

AIR-BRAKE ASSOCIATION.—T. L. Burton, c/o Westinghouse Air Brake Company, Thirty-fourth floor, Empire State Bldg., New York.

ALLIED RAILWAY SUPPLY ASSOCIATION.—F. W. Venton, Crane Company, Chicago.

ASSOCIATION OF AMERICAN RAILROADS.—J. R. Downes, vice-president operations and maintenance department, Transportation Building, Washington, D. C.

DIVISION I.—OPERATING.—SAFETY SECTION.—J. C. Caviston, 30 Vesey street, New York.

DIVISION V.—MECHANICAL.—V. R. Hawthorne, 59 East Van Buren street, Chicago.

COMMITTEE ON RESEARCH.—H. A. Johnson, chairman (Director of Research, Association of American Railroads), Chicago.

DIVISION VI.—PURCHASE AND STORES.—W. J. Farrell, 30 Vesey street, New York.

DIVISION VIII.—MOTOR TRANSPORT.—CAR SERVICE DIVISION.—C. A. Buch, Transportation Building, Washington, D. C.

AMERICAN RAILWAY TOOL FOREMEN'S ASSOCIATION.—G. G. Macina, 11402 Calumet avenue, Chicago.

AMERICAN SOCIETY FOR TESTING MATERIALS.—C. L. Warwick, 260 S. Broad street, Philadelphia, Pa. Thirty-eighth annual meeting, June 24-28, Book-Cadillac Hotel, Detroit, Mich.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS.—C. E. Davies, 29 W. Thirty-ninth street, New York.

RAILROAD DIVISION.—Marion B. Richardson, 192 E. Cedar st., Livingston, N. J. Next meeting, Cincinnati, Ohio, June, 1935.

MACHINE SHOP PRACTICE DIVISION.—G. F. Nordenholt, 330 W. Forty-second st., New York.

MATERIALS HANDLING DIVISION.—M. W. Potts, Alvey-Ferguson Company, 1440 Broadway, New York.

OIL AND GAS POWER DIVISION.—M. J. Reed, 2 W. Forty-fifth St., New York.

FUELS DIVISION.—W. G. Christy, Department of Health Regulation, Court House, Jersey City, N. J.

ASSOCIATION OF RAILWAY ELECTRICAL ENGINEERS.—Jos. A. Andreucetti, C. & N. W., 1519 Daily News Building, 400 W. Madison St., Chicago, Ill.

CANADIAN RAILWAY CLUB.—C. R. Crook, 2276 Wilson avenue, Montreal, Que. Regular meetings, second Monday of each month except in June, July and August at Windsor Hotel, Montreal, Que.

CAR DEPARTMENT OFFICERS ASSOCIATION.—A. S. Sternberg, master car builder, Belt Railway of Chicago, 7926 South Morgan street, Chicago.

CAR FOREMEN'S ASSOCIATION OF CHICAGO.—G. K. Oliver, 2514 West Fifty-fifth street, Chicago. Regular meetings, second Monday in each month except June, July and August, La Salle Hotel, Chicago, Ill.

CAR FOREMAN'S ASSOCIATION OF OMAHA, COUNCIL BLUFFS AND SOUTH OMAHA INTERCHANGE.—E. R. Phillips, car department, Chicago & North Western Railway, Council Bluffs, Iowa. Regular meetings, second Thursday of each month at 1:15 p. m. at Union Pacific shops, Council Bluffs.

CENTRAL RAILWAY CLUB OF BUFFALO.—Mrs. M. D. Reed, Room 1817, Hotel Statler, Buffalo, N. Y. Regular meeting, second Thursday each month except June, July and August at Hotel Statler, Buffalo.

EASTERN CAR FOREMEN'S ASSOCIATION.—E. L. Brown, care of the Baltimore & Ohio, Staten Island, N. Y. Regular meetings, fourth Friday of each month, except June, July, August and September.

INDIANAPOLIS CAR INSPECTION ASSOCIATION.—R. A. Singleton, 822 Big Four building, Indianapolis, Ind. Regular meetings first Monday of each month, except July, August and September, at Hotel Severin, Indianapolis, at 7 p. m.

INTERNATIONAL RAILROAD MASTER BLACKSMITHS' ASSOCIATION.—W. J. Mayer, Michigan Central, 2347 Clark avenue, Detroit, Mich.

INTERNATIONAL RAILWAY FUEL ASSOCIATION.—T. D. Smith, 1660 Old Colony building, Chicago.

INTERNATIONAL RAILWAY GENERAL FOREMEN'S ASSOCIATION.—William Hall, 1061 W. Wabasha street, Winona, Minn.

MASTER BOILERMAKERS' ASSOCIATION.—A. F. Stiglmeier, secretary, 29 Parkwood street, Albany, N. Y.

NEW ENGLAND RAILROAD CLUB.—W. E. Cade, Jr., 683 Atlantic avenue, Boston, Mass. Regular meeting, second Tuesday in each month, excepting June, July, August and September.

NEW YORK RAILROAD CLUB.—D. W. Pye, Room 527, 30 Church street, New York. Meetings, third Friday in each month, except June, July and August, at 29 West Thirty-ninth street, New York.

NORTHWEST CAR MEN'S ASSOCIATION.—E. N. Myers, chief interchange inspector, Minnesota Transfer Railway, St. Paul, Minn. Meeting first Monday each month, except June, July and August, at Minnesota Transfer Y. M. C. A. Gymnasium building, St. Paul.

PACIFIC RAILWAY CLUB.—William S. Wollner, P. O. Box 3275, San Francisco, Cal. Regular meetings, second Thursday of each month in San Francisco and Oakland, Cal., alternately.

RAILWAY CLUB OF GREENVILLE.—Ralph D. Stewart, 21 Sherrard avenue, Greenville, Pa. Regular meeting third Thursday in month, except June, July and August.

RAILWAY CLUB OF PITTSBURGH.—J. D. Conway, 1941 Oliver building, Pittsburgh, Pa. Regular meeting fourth Thursday in month, except June, July and August, Ft. Pitt Hotel, Pittsburgh, Pa.

RAILWAY FIRE PROTECTION ASSOCIATION.—R. R. Hackett, Baltimore & Ohio, Baltimore, Md.

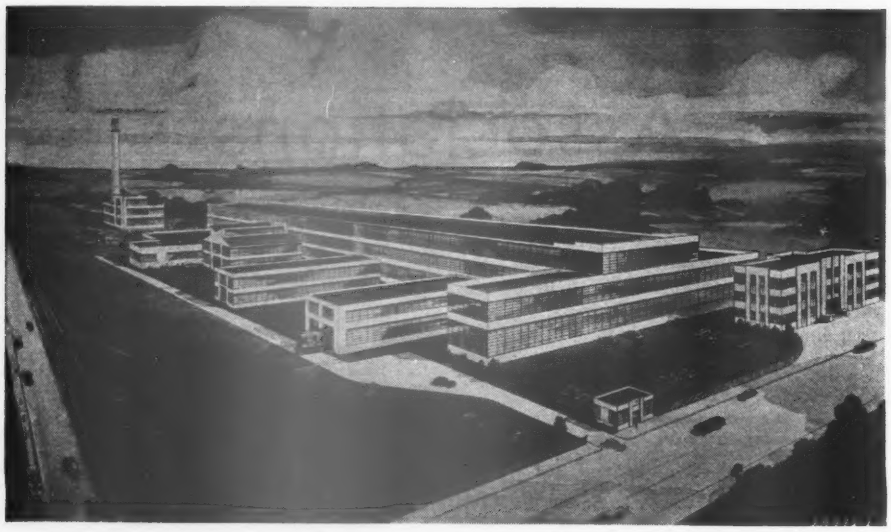
RAILWAY SUPPLY MANUFACTURERS' ASSOCIATION.—J. D. Conway, 1841 Oliver building, Pittsburgh, Pa. Meets with Mechanical Division and Purchases and Stores Division, Association of American Railroads.

SOUTHERN AND SOUTHWESTERN RAILWAY CLUB.—A. T. Miller, P. O. Box 1205, Atlanta, Ga. Regular meetings third Thursday in January, March, May, July and September. Annual meeting, third Thursday in November, Ansley Hotel, Atlanta, Ga.

TORONTO RAILWAY CLUB.—R. H. Burgess, Box 8, Terminal A, Toronto, Ont. Meetings, first Friday of each month except June, July, August and September.

TRAVELING ENGINEER'S ASSOCIATION.—W. O. Thompson, 1177 East Ninety-eighth street, Cleveland, Ohio.

WESTERN RAILWAY CLUB.—C. L. Emerson, 822 Straus building, Chicago. Regular meetings third Monday in each month except June, July, August and September.



DIESEL-ELECTRIC LOCOMOTIVE PLANT TO BE BUILT AT MCCOOK, ILL., FOR ELECTRO-MOTIVE CORPORATION.—This will be the first complete plant for the exclusive manufacture of Diesel-electric locomotives in the country. The contract for its erection has been let to the Austin Company, Cleveland, Ohio. The various portions of the project include a three-story main office building, 40 ft. by 140 ft.; an employment office, 15 ft. by 30 ft.; an erecting and machine shop, 170 ft. by 550 ft.; a blacksmith shop, 70 ft. by 75 ft.; annealing ovens, 22 ft. by 200 ft.; a sand blast building, 30 ft. by 150 ft.; a paint shop, 50 ft. by 140 ft.; a warehouse, 48 ft. by 140 ft.; a power house, 40 ft. by 60 ft., and railroad trackage, track scales and roads. The main erecting aisle of the erecting and machine shop building will have a 104-ft. clear span 49 ft. high. It will be served by one 200-ton and two 30-ton electric traveling cranes. Two 20-ton cranes will be provided in the machine shop.

NEWS

THE CHICAGO, ST. PAUL, MINNEAPOLIS & OMAHA is making plans to rebuild its 15-stall enginehouse at Itasca, Wis., which was recently damaged by fire. It is estimated that the repair work will cost about $50,000.

Government Report on Dangers of Arch-Bar Trucks

THE Interstate Commerce Commission, speaking through its Bureau of Safety, W. J. Patterson, director, recommends to all railroads that steps be taken at once to inaugurate a program which will definitely eliminate the use of arch-bar trucks from service.

This recommendation is issued in connection with a report on the derailment of a passenger train on the Southern, at Charlotte, N. C., on January 13, when two persons were killed and eight were injured. Warnings of the necessity of eliminating arch-bar trucks have been given in connection with a number of accident reports lately, mention being made, in each case, of the rule adopted by the American Railway Association (now Association of American Railroads) that these trucks should not be offered in interchange after a certain date (this date has been repeatedly postponed and lately has been changed from January 1, 1936, to January 1, 1938). The present report, however, is predicated on the failure of such a truck on a tender; and the operation of tenders is not affected by the interchange rules. On the Southern, it was found that of 1,932 tender trucks in service, 1,307 were of the arch-bar type. Of these, 265 were in passenger service.

The accident occurred on southbound passenger train No. 31, moving at about 50 miles an hour, which was derailed on a curve of 6 deg.; superelevation 2¾ in. The locomotive was overturned and one car fell down a bank into a street. Of several breaks in the arch-bars (one truck being under the tender and one under a baggage car) all were apparently new, with no evidence of old flaws or cracks. The trucks had been repaired, but there was no precise information as to what in them was new material. The track on this curve was not well-maintained, the elevation and the gage both being uneven, but the conclusion is that the failure of a truck was the main cause.

$1,500,000 Sought by C. & N. W. for Equipment Repairs

THE Chicago & North Western has filed the first application for an equipment loan under the loaning powers recently conferred upon the Reconstruction Finance Corporation and the Interstate Commerce Commission to aid in the financing of railroads for purposes of reorganization, consolidations, maintenance, construction, or purchase of equipment. The application of the C. & N. W. is for improvement and repairs to equipment, including improvements and general repairs to 12 locomotives, classified repairs to 95 locomotives, air conditioning and modernizing 66 passenger cars, modernizing 4 dining cars and 2 lounge cars, and installing Evans auto-loaders on 200 cars.

"The Development of the Lathe"

A NEW industrial film entitled "The Development of the Lathe" has been produced by the Monarch Machine Tool Company, Sidney, Ohio. The film pictures the original tree or pole lathe of 500 A. D. through its successive stages of development; the first wood screw cutting lathe of the fifteenth century; the first iron lathe of 1800 A. D., the first quick-change gear lathe of 1890, and the first geared head lathe of 1900. A number of Monarch machines are then shown in operation. These films are available, in two sizes, for loan to industrial organizations, technical societies, educational institutions, etc., through W. E. Whipp of the Monarch Company.

New Equipment

CAR ORDERS			
Purchaser	No. of cars	Type of car	Builder
C. B. & Q.	1[1]	4-car articul. train	Edw. G. Budd Mfg. Co.
CAR INQUIRIES			
A. T. & S. F.	2	75-ft. streamline mail and bagg.	....
U. S. Engineer Office[2]	1	Flat	....
LOCOMOTIVE ORDERS			
Road	No. of locos.	Type of loco.	Builder
S. A. L.	4	16,000 gal. tenders	Baldwin Loco. Wks.
W. & L. E.	8	0-6-0 switch.	Co. shops at Brewster, Ohio
LOCOMOTIVE INQUIRIES			
L. & A.	4	Mikado	....
Russian Government[3]	..	8-wheel switchers	....
MISCELLANEOUS ORDERS			
Road	Type of equip.	For use on	Order placed with
N. Y. C.	Bearings, main and side rods and other reciprocating parts	"Commodore Vanderbilt"[4]	Timken Roller Bear. Co.
Western Md.	Loco-Valve pilots	5 frt. locos.	Valve Pilot Corp.

[1] This streamline Zephyr train No. 4 will operate between St. Louis, Mo., and Burlington, Iowa.
[2] Rock Island, Ill.
[3] Inquiry through Amtorg Trading Corp.
[4] This locomotive is now equipped with Timken bearings on all axles.

N. Y. C. Streamlined Locomotive in Regular Service

The New York Central's streamlined steam locomotive—the Commodore Vanderbilt—made its first run in regular service on February 19 when it hauled the first section of the Twentieth Century Limited from Chicago to Toledo, Ohio, 233 miles. On February 20 it hauled the westbound Century from Toledo to Chicago and will continue on this assignment for a time.

Letter Ballot on Materials

The revised material specifications proposed by the Association of American Railroads, Mechanical division, committee on this subject, were submitted to the members of the division in a letter ballot, the results of which have just been made available. The recommendations were divided into 41 separate propositions, all of which were approved and will be included in the Manual of Standard and Recommended Practice.

The first proposition pertained to the issuance and numbering of material specifications and the second to the elimination of nine separate specifications, as recommended by the committee, for such materials as steel axles, shafts, bolts and nuts, cylinder parts, fire hose, engine-bolt iron, foundry pig iron, axle light belting, welding wire, etc. In the Type-E coupler specifications, the part covering material was ordered removed and specifications for carbon steel castings substituted as recommended by the committee. Further propositions pertained to the adoption of revised specifications for carbon steel, car axles, forgings, billets, tires, boiler tubes, boiler steel, steel structural shapes, carbon steel castings, springs, chain, bronze bearings, air-brake and signal hose, steam and hot water hose, rubber goods, etc.

High-Speed Train Operations

Union Pacific.—During the first month of operation, the three-car articulated stream-lined train of the Union Pacific was filled to capacity on practically every run, while at times it carried twice as many people as there were seats. During February, 6,505 persons rode the train, 3,062 being carried eastbound and 3,443 westbound. On one trip, a total of 254 persons patronized the train while on February 24, the train entered Kansas City with 233 persons on board. On the eastbound trips during February, an average of 109 persons rode the train, while westbound the average was 123.

Because of the popularity of the streamliner, an additional run from Kansas City to Topeka, Kan., was added on February 23. The train leaves Salina, Kan., at 7 a. m., arrives in Kansas City at 10.30 a. m. and at 11 a. m. starts on its new run to Topeka, where it arrives at 12:08 p. m. It leaves Topeka at 12:30 p. m., arrives in Kansas City at 1:38 p. m. and leaves that city at 4 p. m. for Salina, where it arrives at 7:30 p. m.

Chicago & North Western.—The running time of the C. & N. W. "400" express train between Chicago and the Twin Cities will be reduced from 7 hr. to 6½ hr. on April 28. Under the new schedule, 5 min. of the cut will be absorbed in the run from Chicago to Milwaukee, where the 85 miles will be covered in 75 min., or at the rate of 68 m.p.h., compared with the present average of 63.8. The remaining 25 min. will be absorbed in the schedule between Milwaukee and St. Paul, where the new schedule will call for an average speed of 63 m.p.h., as compared with the present 57. This decision to reduce running time is based upon the success of the train in meeting its schedule since its inauguration on January 2. On numerous occasions, the train has had no difficulty in making up delays. An outstanding example occurred during the first part of March when the southbound train covered the 8 miles between Highland Park, Ill., and Indian Hill in 5 min., or at an average of 96 m.p.h.

Progress in Air-Conditioning Programs

Road	No. of cars	Type of car	Type of system	Builder
C. & O.	14	Coaches	....	Pullman-Std. Car Mfg. Co.
C. & E. I.	30	Parlor, sleep, and dining	....	....
C. R. I. & P.	1[1]	Parlor	Thermo-gravity	American C. & F. Co.
N. Y. C. & St. L. ...	1	Coach	....	Pullman-Std. Car Mfg. Co.
N. Y. C.	7[2]	Dining	Electro-mech.	....
	46[2]	Coaches		
	182[2]	Pullman	Pull-speed control	Pullman-Std. Car Mfg. Co.
N. & W.	4[3]	Pass.	....	....
	4[3]	Dining	....	....
S. A. L.	5	Coaches	....	Pullman-Std. Car Mfg. Co.

[1] One set of 7 ton's capacity.

[2] When this work is completed the N. Y. C. will be operating 537 cars completely air-conditioned. The 7 dining cars are being equipped in its own shops at West Albany, N. Y.; the 46 coaches, in its shops at Beech Grove, Ind., and the 182 Pullman cars, at the Pullman shops. When these cars are ready all the road's principal trains operating east and west between the principal cities they serve will be air conditioned.

[3] The Norfolk & Western will spend approximately $80,000 for the air-conditioning of eight additional units of its standard passenger equipment, the work to be done in its Roanoke, Va., shops. The project calls for the air conditioning and remodeling of four standard passenger coaches and the air conditioning of four dining cars. In addition to air conditioning, the remodeling of the four coaches will include the construction of new arched ceilings, the installation of dome lights, new rubber tile floors and repainting the interior of the cars. The company was reported in the March *Railway Mechanical Engineer* as having started similar work costing $80,000 on the air conditioning and remodeling of eight passenger coaches. Work on the new job will be carried on concurrently with the cars now being air conditioned in the local shops and it is expected that all 16 units will be placed in service in June. The completion of this work will give the Norfolk & Western 62 air-conditioned units of passenger equipment; these include 30 coaches, 11 dining cars and 21 Pullman cars.

North Coast Limited Cars To Be Modernized

Modern air-conditioned coaches will be added to the North Coast Limited of the Northern Pacific for the coming tourist season. Not only will the coaches be air-conditioned and roller-bearing equipped, but the interior arrangement and appearance of the cars will suggest luxury and refinement. The main room will be entirely enclosed with bulkheads. A lounging and dressing room for men, with a seating capacity for 10 in arm-type chairs upholstered in leather, will be at one end of the car, while at the other end will be a ladies' lounge and separate dressing room with a seating capacity for 8, provided with a davenport for 4, individual chairs and double seats. The floor will be carpeted and the ladies' dressing room will have several mirrors, including one full-length mirror at the door.

The main room in the new cars will seat 36 to 40 persons. Some coaches will be decorated in brown, others in blue and in gray, adding a variety in color scheme. The lighting will be of the over-head type and, in addition, individual lights will be above each double seat, in direct control of the passenger. The seats are of new design, 44½ in. wide with the back and seat cushions divided in the center.

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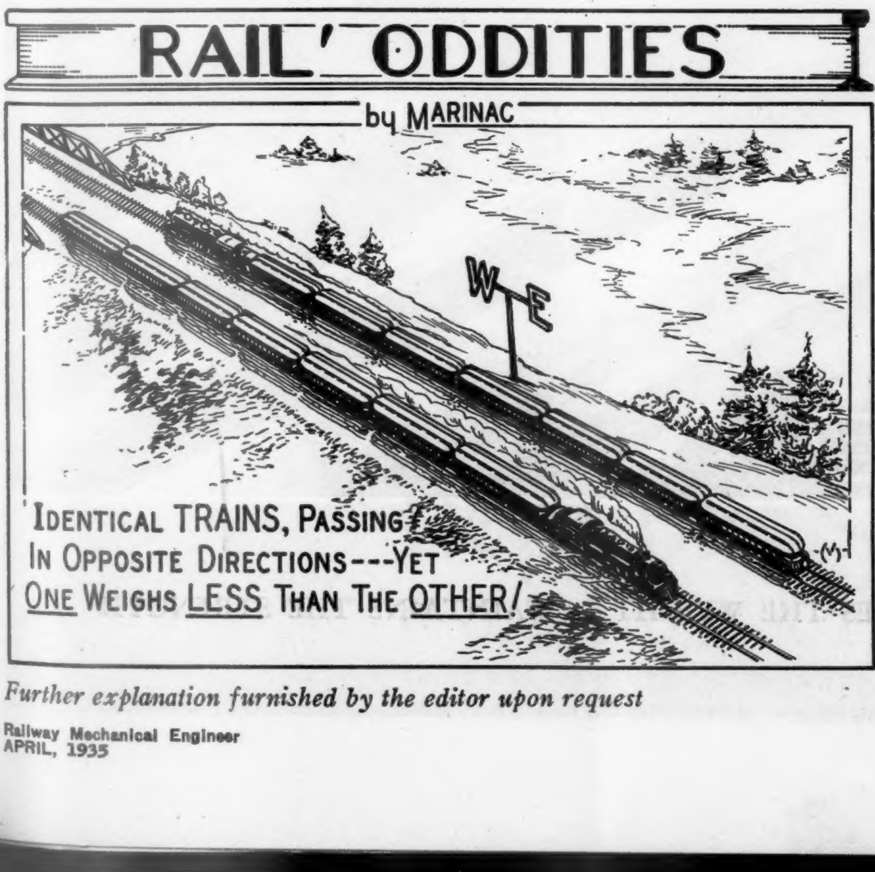

Further explanation furnished by the editor upon request

These Steels are

REPUBLIC STEEL

REDUCES THE WEIGHT . . . MAINTAINS THE STRENGTH

well named . . .

REPUBLIC Double Strength

	Ordinary Carbon Steel	Republic Double Strength Grade 1	Republic Double Strength Grade 1-A
Yield Point	35,000 lbs. p.s.i.	60,000 lbs. p.s.i. min.	70,000 lbs. p.s.i. min.
Tensile Strength	50,000 lbs. p.s.i.	75,000 lbs. p.s.i. min.	90,000 lbs. p.s.i. min.
Elongation in 2 in.	25%	25% min.	18% min.

A moment's comparison of the physical properties of Republic Double Strength Steels and those of ordinary carbon steel is, itself, justification of the name and proof that one pound of steel can be made to do the work of two.

Here are two comparatively new steels that permit the use of 30% to 40% lighter sheets, strip or plates without sacrifice of strength or of safety factor—that make possible a reduction of dead weight in transportation equipment and increase revenue load capacity proportionately.

Cold forming tests have shown the superiority of Republic Double Strength Steels from the standpoint of ductility over similar products with the same physicals. Tempering increases their yield points 20,000 to 30,000 pounds over the figures shown here. They can be arc, gas or spot welded with the least change in ductility at or adjacent to the weld. There is no air hardening, hence more safety in fabrication. And because of their copper, nickel and molybdenum alloy content, they will resist corrosion far better than ordinary carbon steels and show a life equal to or greater than that of the heavier sections which have been standard practice in the past.

These steels are destined to play an increasingly important part in the construction of new transportation equipment and in the modernizing of old. We shall be glad to furnish more detailed information on request and to make recommendations covering their use for specific applications.

REPUBLIC STEEL

Republic Steel CORPORATION

GENERAL OFFICES · · · YOUNGSTOWN, OHIO
CENTRAL ALLOY DIVISION · · MASSILLON, OHIO

Supply Trade Notes

W. R. Walsh

W. R. Walsh, who has been elected vice-president of the Ewald Iron Company, with headquarters at Chicago, after attending the University of Illinois, began his business career in 1917 as an office and sales department employee of the Standard Oil Company. Mr. Walsh entered the railway supply business in 1920 and for several years represented the Glidden Company and several other supply companies. In 1926 he was appointed representative of the Ewald Iron Company and in 1928 became resident sales manager.

The Ex-Cell-O Aircraft & Tool Corporation, Detroit, Mich., manufacturers of hardened and ground bushings and pins for locomotive and passenger-car work, has appointed the following companies as its representatives in the railroad field: Modern Supply Company, 53 W. Jackson boulevard, Chicago; C. E. Murphy, 415 Midland building, Cleveland, Ohio; Geo. A. Secor, 4485 Duncan avenue, St. Louis, Mo.; W. W. Weller, 426 Park Square building, Boston, Mass.; W. Searls Rose, 69 Dey street, New York; Scott Donahue, 858 Graybar building, New York; W. J. Church, 707 Queen and Crescent building, New Orleans, La.; W. H. Erskine, Box 72 Traffic Station, Minneapolis, Minn.; A. R. Sleath, 502 Haverford avenue, Narberth (Philadelphia), Pa.; D. E. McCulley, 1101 Jackson street, Omaha, Neb., and W. D. Otter, 234 Rialto building, San Francisco, Cal.

C. H. Rhodes, assistant general manager of sales of the bar, semi-finished and alloy steel division of the Illinois Steel Company, with headquarters at Chicago, has been appointed director of purchases of the United States Steel Corporation, with headquarters at New York. He has been succeeded by Griswold A. Price, who since January 1 has been manager of sales at St. Louis, Mo., for the Carnegie Steel Company, the Illinois Steel Company and the Tennessee Coal, Iron & Railroad Company. Robert Corson, Jr., succeeds Mr. Price at St. Louis. William B. Weston, manager of sales in the Detroit district for the Carnegie Steel Company, the Illinois Steel Company and the Tennessee Coal, Iron & Railroad Company, has been appointed assistant to the vice-president and general manager of sales of the Carnegie Steel Company, with headquarters at Pittsburgh, Pa., and has been succeeded by Philip M. Guba, assistant manager, at Detroit, Mich.

The Safety Car Heating & Lighting Company has moved its St. Louis office to 915 Olive street. S. I. Hopkins is manager.

Erle G. Hill has been appointed director of research of the Lukens Steel Company, Coatesville, Pa. Since 1930 Mr. Hill has had a research fellowship at the Mellon Institute of Industrial Research.

The Atlas Supply Company, Inc., 35 Woodward avenue, Brooklyn, N. Y., has been appointed warehouse distributors of rust-resisting Toncan Iron sheets of the Republic Steel Corporation, Youngstown, Ohio.

Frank P. Roesch, who has been elected a vice-president of the Standard Stoker Company, Inc., with headquarters at Chicago, was born on April 14, 1864, in Alsace, and came to the United States immediately after the close of the Franco-Prussian war. Following his graduation from high school he entered railway service in 1877 as machinist apprentice at Trenton, Mo., on the Chicago, Rock Island & Pacific. He subsequently served in several railroad shops in various positions, including that of enginehouse foreman, also as a locomotive engineman. In 1891 he was appointed traveling engineer on the Denver & New Orleans, now the Colorado & Southern, and later became general traveling engineer. In 1901 he was appointed master mechanic of the Colorado & Southern and subsequently served as master mechanic on the Chicago & Alton, the Southern, and the El Paso & Southwestern, now part of the Southern Pacific. Mr. Roesch served from 1918 to 1920 as regional fuel supervisor, Northwest region, U. S. R. A. At the termination of federal control he became western manager of the Standard Stoker Company, Inc., and in 1923 was appointed general sales manager. Mr. Roesch has been a frequent contributor to various railroad journals since 1898. He is an authority on subjects pertaining to locomotive operation, combustion, etc., and is a past president of the Traveling Engineers' Association. He has also been active in the affairs of the International Railway Fuel Association and the Smoke Prevention Association. He is a member of a large number of other technical and mechanical organizations and the patentee of a number of stoker appliances.

Frank P. Roesch

J. C. Kaimer has become engineering and district sales manager of the Apex Tool & Cutter Company, Shelton, Conn., servicing inserted tools in New York, New Jersey and Pennsylvania territory. Mr. Kaimer for the past twelve years had been engineer and representative of the O. K. Tool Company.

S. H. Blackwood, assistant district sales representative in the New York territory of the Reading Iron Company, Philadelphia, Pa., has been appointed district sales representative in the southern territory, with headquarters at 1104 Continental building, Baltimore, Md. Mr. Blackwood takes the position held for many years by W. J. White, who recently resigned. W. N. Johnson has been appointed salesman in the New York territory.

John M. Mulholand, special representative of railroad sales at Chicago, for the Youngstown Sheet & Tube Company, Youngstown, Ohio, has been appointed manager of railroad sales for the company. Mr. Mulholand's headquarters will continue to be at Chicago. He was born at Pittston, Pa., and after attending high school in that state, was a member of the class of 1910, University of Michigan, marine engineering course. From 1910 to 1917 Mr. Mulholand followed engineering, principally mining work, and then served in the war, entering the service through the Officers Training School Tank Corps. From the close of the war to 1932 he was actively engaged in the railroad equipment field, first as district sales manager of Mudge & Company, and then as vice-president of sales of the O. F. Jordan Company. Mr. Mulholand has been with the Youngstown Sheet & Tube Company since December, 1932.

John M. Mulholand

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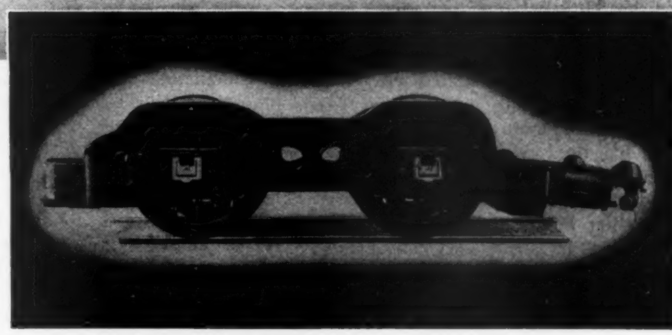

HEAVY TRAINS

Move faster and smoother with the BOOSTER

On the streamlined Pacifics for the Boston and Maine recently delivered by Lima Locomotive Works, Incorporated, The Locomotive Booster was incorporated in the original design.

Addition of The Booster increases the tractive effort in starting, in accelerating to road speeds and over the hard pulls from 40,900 pounds to 52,800 pounds.

This extra tractive effort makes for smooth starting and quick acceleration. It materially aids in maintaining fast schedules.

Wherever passenger comfort with heavy trains and high speeds is involved, The Locomotive Booster is a valuable operating factor. It capitalizes idle weight and spare steam.

All replacement parts furnished by Franklin Railway Supply Company are identical as to materials, design, clearances and workmanship with the parts they replace. They guarantee the same unfailing reliability of service.

FRANKLIN RAILWAY SUPPLY COMPANY, INC.

NEW YORK CHICAGO MONTREAL

THE NATIONAL COPPER PAINT COMPANY, Chicago, has been formed to manufacture and market liquid copper paint. Officers of the company are: President, H. M. Rice, manager of the Nichols Copper Company, Chicago, a unit of the Phelps-Dodge Corporation; executive vice-president, C. L. Welch; and secretary-treasurer, Frederick A. McLauchlan, president of George B. Carpenter & Co.

KNUT NORDENSON, for 20 years chief designer of the McIntosh & Seymour Corporation, at Auburn, N. Y., has been appointed chief engineer of the company to fill the vacancy caused by the resignation of Paul A. Ritter, Sr., who had been chief engineer for 10 years. Mr. Nordenson received his technical education at Tekniska Hogskolan, Stockholm, Sweden. Previous to 1915 he was designer for eight years at Aktiebolaget Atlas Diesel, Stockholm, and was associated also with a number of other oil engine companies. Karl Volmar Anderson, who for the past five years has been responsible for the design of the McIntosh & Seymour high-speed and railway type engines, has been appointed chief designer for the company. Mr. Anderson has had experience with the Cummins Engine Company, Columbus, Ind., as a designer, as well as with other Diesel engine companies both in this country and in Europe. James T. Lewis, chief inspector of the corporation, will continue in that position and will also be in charge of field service of Diesel engines. William N. Nichols, who has been in charge of field service for the past eight years, is now in charge of all experiments in combustion and all engine testing.

BLAIR C. HANNA, who has been appointed manager of sales of the Ralston Steel Car Company, Columbus, Ohio, was born in Pittsburgh, Pa., and began his business career in 1899 with the Pressed

Blair C. Hanna

Steel Car Company. In 1908 he was appointed chief estimator of the Ralston Steel Car Company, and in 1912 became assistant to the vice-president. During 1918 and 1919 he was in military service, serving with the Department of Military Railways of the chief of engineers office at Washington. On the completion of his military duties he was appointed chief engineer of the Ralston Co. In 1920, he was appointed also assistant to the president.

F. A. Livingston

F. A. LIVINGSTON, who has been elected president and general manager of the Ralston Steel Car Company, Columbus, Ohio, has been associated with the company since January 1, 1906, when he entered its employ as secretary to the president. After holding the positions of bookkeeper and paymaster, he was appointed assistant secretary and assistant treasurer on June 7, 1911. On August 1, 1916, he was elected a director; on February 9, 1917, secretary and treasurer, and on September 15, 1923, vice-president and treasurer. Mr. Livingston is also president of the Mifflin Realty Company and the Ralston Scales Corp.

ERLE G. HILL has become associated with the Lukens Steel Company, Coatesville, Pa., as director of research. Mr. Hill has for a number of years been employed as a metallurgist, both in instruction and research at several universities.

THE AMERICAN LOCOMOTIVE COMPANY has appointed the Pacific Car & Foundry Company of Seattle, Wash., as its sales representatives for the states of Washington, Oregon and Idaho, succeeding the Zimmerman-Wells-Brown Co. of Portland.

Obituary

WILLIAM L. REID, vice-president of the Lima Locomotive Works, Inc., died suddenly, on March 9, at Lima, Ohio. Mr.

William L. Reid

Reid was a native of Paterson, N. J., and was well known in railroad circles. He started his career as an apprentice in the old Rogers Locomotive Works at Paterson, N. J. In 1901 he became general works manager of the American Locomotive Company, and in 1918 was elected vice-president in charge of manufacturing of the Lima Locomotive Works, Inc. Mr. Reid spent his entire business life in locomotive development and was an active member of the old American Railway Master Mechanics Association.

RALPH H. CLORE, general sales manager of The Medart Company, St. Louis, Mo., died on March 6.

GRANT WARREN SPEAR, retired vice-president of the Dearborn Chemical Company, at New York, and for years its eastern manager, died at his home in Palm Beach, Fla., on March 22.

JOHN S. KEEFE, who retired from the presidency of the American Steel & Wire Company on January 1, 1933, died on March 3 at his home, Oak Park, Ill. Mr. Keefe was born on January 24, 1864, at Boston, Mass.

DR. WM. H. BEARDSLEY, a director of the Jones & Lamson Machine Company, Springfield, Vt., died on March 2. Dr. Beardsley, who was born in 1881, was a graduate of Yale Medical School and had considerable medical experience. He became manager of the Fay automatic lathe department of the Jones & Lamson Machine Company in 1915, later becoming production superintendent and a director.

WILLIAM F. WALSH, manager of the railroad sales department of the Standard Oil Company of New Jersey, died on March 19 after a brief illness at the New York Hospital, New York City. Mr. Walsh was born at Cleveland, Ohio, in 1882, where he spent his early life, attending grammar school and St. Ignatius College, afterwards serving an apprenticeship on the Norfolk & Western. He subsequently entered the mechanical department

William F. Walsh

of the Chesapeake & Ohio, serving in various official capacities until he became supervisor of air brakes, from which position he resigned in 1912 to become lubrication engineer in the mechanical department of the Galena Signal Oil Company. In 1917 he entered the service of the United States army, being commissioned a captain. Following the war he returned to the Galena Signal Oil Company, resigning in 1919 to become assistant superintendent of motive power of the Chicago, Milwaukee & St. Paul, now the Chicago,

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Milwaukee, St. Paul & Pacific. In 1923 he returned to the Galena Company as supervising engineer in charge of the Western territory, being located in Chicago until 1925, at which time he was promoted to manager of the St. Louis office. In 1926 he went with the Standard Oil Company of New Jersey to organize and supervise its railroad sales department. Mr. Walsh was a member of a number of organizations, including the American Society of Mechanical Engineers, the New York Railroad Club, the Traveling Engineers' Association, the International Railway Fuel Association and the Brotherhood of Locomotive Engineers.

Joel S. Coffin Dies

Played an important part in the development of the modern steam locomotive

Joel Stephen Coffin died at Miami Beach, Fla., on Monday, March 11. Born in a country district in Michigan, in 1861, and coming from a family of pioneers, it was not surprising that he developed into a strong individualist and an aggressive business leader. With such a background, tempered by a spirit of tact and diplomacy, he forged steadily ahead, first in the mechanical and operating departments of the railroads and then in the railway supply industry, to the extent that for the past quarter of a century, at least, he has been regarded as one of the most constructive and progressive leaders in the railway supply industry.

The companies with which he has been associated have been characterized by their activities in research, in the development of new designs and appurtenances, and for their ability in servicing and promoting their various products. The servicing of these products has not only insured the railroads using them more effectively, but it has made it possible to follow their operation critically and insure their improvement and development as conditions became more severe or required modifications in the design. Mr. Coffin also had the gift of selecting and developing executives and engineering experts of unusual ability. It is not surprising, therefore, that he should have been so important a factor in the development of the modern steam locomotive, which at the beginning of the century some of the over-enthusiastic advocates of electrification prophesied would be superseded within a decade or two. His influence in constructive accomplishment in the railroad field will not be lost, but will be felt as long as the steam locomotive exists, and, judging from present indications, its days are not yet numbered.

Joel S. Coffin

Mr. Coffin was born in 1861 in Wales Township, St. Clair County, Mich., and his boyhood was spent in Elm Hall, Mich. At 18 years of age Mr. Coffin became a machinist apprentice in the shops of the Chicago & West Michigan at Muskegon, Mich. He left that occupation in order to become a locomotive fireman and later a locomotive engineer on the same road. In 1885 he entered the service of the Wisconsin Central as a locomotive engineer, and in 1890 was promoted to the position of road foreman of engines. In 1892 he became a member of the mechanical expert staff of the Galena Signal Oil Company. He became manager of that department in 1895 and in 1907 was elected vice-president. He resigned in 1909 to become a vice-president of the American Brake Shoe & Foundry Company, with headquarters in New York.

Meanwhile, in 1902, while with Galena Company, he, with Samuel G. Allen, organized the Franklin Railway Supply Company. In 1910, while with the American Brake Shoe & Foundry Company, Mr. Coffin organized the American Arch Company. He also took an active part in the development of the Locomotive Superheater Company. In 1911 he resigned from the American Brake Shoe & Foundry Company, to devote his entire time to the Franklin Railway Supply Company, the American Arch Company, the Locomotive Superheater Company, and other companies which later became associated with these organizations. In 1916 he became chairman of the board of the Lima Locomotive Works, Inc. At the time of his death Mr. Coffin was chairman of the board of the American Arch Company, the Franklin Railway Supply Company and the Lima Locomotive Works, Inc. He was chairman of the executive committee of the Superheater Company and a director of the American Brake Shoe & Foundry Company, G. M. Basford Company, Locomotive Feedwater Heater Company, Rome Iron Mills, Inc., Balmar Manufacturing Company, and other organizations. Mr. Coffin took great pride in retaining his membership in the Brotherhood of Locomotive Engineers. He lived in Englewood, N. J., and took a keen interest in civic affairs. He is survived by his widow and two sons, Charles William Floyd Coffin, vice-president of the Franklin Railway Supply Company, and Joel Stephen Coffin, president of the J. S. Coffin, Jr., Company.

Personal Mention

General

Howard Pyle has been appointed mechanical supervisor in charge of the mechanical department of the Maryland & Pennsylvania at Baltimore, Md., succeeding the late C. L. Adair, master mechanic.

A. Rau, enginehouse foreman of the Northern Pacific at Portland, Oregon, has been transferred to the Argo shops, Seattle, Wash., where, as mechanical foreman, he will have charge of locomotive and car departments.

Master Mechanics and Road Foremen

E. G. Bowie, master mechanic of the Saskatchewan district of the Canadian Pacific at Moose Jaw, Sask., has been transferred to the British Columbia district, with headquarters at Vancouver, B. C.

J. W. McKinnon, locomotive foreman of the Canadian Pacific at Winnipeg, Man., has been promoted to the position of master mechanic at Calgary, Alta., to succeed J. P. Kelly.

E. G. Bowie, master mechanic of the Saskatchewan district of the Canadian Pacific, with headquarters at Moose Jaw, has been appointed master mechanic of the British Columbia district.

J. P. Kelly, master mechanic of the Calgary division of the Canadian Pacific at Calgary, Alta., has been appointed master mechanic of the Saskatchewan district, with headquarters at Moose Jaw, Sask.

Shop and Enginehouse

A. C. Colville has been appointed superintendent of shops on the Great Northern at Hillyard, Wash., succeeding J. M. Hurley, deceased.

William S. Eyerley, Jr., assistant superintendent of the Mount Clare shops of the Baltimore & Ohio, has been appointed superintendent of these shops, with headquarters at Baltimore, Md., succeeding the late T. R. Stewart.

Purchasing and Stores

F. J. McGuinness, division store keeper for the Delaware & Hudson, with headquarters at Oneonta, N. Y., has been appointed superintendent of stores, with headquarters at Colonie, N. Y., succeeding F. C. Reardon, deceased.

T. M. Hawkins, master mechanic for the Quebec Central at Sherbrooke, Que., has been appointed storekeeper, with the same headquarters, succeeding G. H. Mulvagh, who has been transferred to the general office staff. The position of master mechanic has been abolished.

Obituary

J. M. Hurley, superintendent of shops on the Great Northern at Hillyard, Wash., died on February 23.